MODERN PRACTICE IN HEAT ENGINES

TELFORD PETRIE

MODERN PRACTICE IN HEAT ENGINES

MODERN PRACTICE IN HEAT ENGINES

BY
TELFORD PETRIE
M.Sc., Assoc.M.Inst.C.E., A.M.I.Mech.E.
LECTURER IN MECHANICAL ENGINEERING IN THE VICTORIA UNIVERSITY OF MANCHESTER, AND IN THE MANCHESTER MUNICIPAL COLLEGE OF TECHNOLOGY

WITH 111 ILLUSTRATIONS AND 2 FOLDING PLATES

LONGMANS, GREEN AND CO.
39 PATERNOSTER ROW, LONDON, E.C.4
55 FIFTH AVENUE, NEW YORK
BOMBAY, CALCUTTA AND MADRAS
1922

PREFACE

A RESIDENT engineer at one of our experimental sea stations during the war, who was an expert in what Kipling once called the "Bridge-Dam-Girder" line, was present at a committee meeting when the silent propulsion of ships was under discussion. He said: "Gentlemen, at our station experiments are being carried out on a new type of engine, which is so silent that it is called the 'still engine.'" If this book ever reaches that engineer's library, I shall feel that it has not been written altogether in vain. I may even go further and express the hope that the steam section will interest the internal-combustion engineer, and *vice versâ*.

The steam turbine section is largely based, by permission, on the lectures of my chief, Professor Gerald Stoney, D.Sc., F.R.S., and some of the descriptive part in this section, which is appearing in our article on "The Physics of the Steam Turbine" in the "Dictionary of Applied Physics," is used here through the courtesy of Messrs. Macmillan & Co., Ltd.

I am also grateful to my former colleague, W. J. Walker, Ph.D., who has now gone to St. Andrew's University, for the free use he has allowed me to make of his various investigations into the fundamental theory of the internal-combustion engine. He has also kindly undertaken the thankless task of reading through the proofs of this section.

A book of this description could not have been written without the willing collaboration of a large number of engineering firms and societies. Although their contributions have been acknowledged where they occur in the text, I should like to place on record here my indebtedness to the unfailing courtesy which I have always received.

The technical press has also been most generous in allowing me to reproduce drawings which they have published. In this connection I would especially thank the Editors of *The Engineer*, *Engineering*, *The Power User*, and the *Zeitschrift des Vereins deutscher Ingenieure.*

In conclusion, I would also express my gratitude to S. Lees, Dempster Smith, S. Plews, T. Bevan, T. H. Jones, Miss M. A. Booth, and others, as well as third-year students in the Mechanical Engineering Department here, for the various forms of help which they have contributed.

TELFORD PETRIE.

MUNICIPAL COLLEGE OF TECHNOLOGY,
SACKVILLE STREET,
MANCHESTER.
February, 1922.

CONTENTS

SECTION I

STEAM BOILERS

SECTION II

STEAM PRIME MOVERS

SECTION III

INTERNAL-COMBUSTION ENGINES

INDEX TO ILLUSTRATIONS

[*The numbers refer to pages.*]

FOLDING PLATES

SECTION I

STEAM BOILERS

CHAPTER I

DEFINITIONS AND THERMAL EFFICIENCY

General Considerations.—The oldest method of obtaining mechanical power from heat is through the agency of steam generated in a boiler.

The object of the boiler is to convert the heat energy latent in cold fuel into the heat energy contained in steam, so that some of it may eventually be used to do useful work through the medium of a steam engine or of a steam turbine.

In the process of raising steam from water there are broadly two stages. The fuel is first burned in a furnace to convert it into flame and a mixture of hot gases. Secondly, the flame and gases are brought into contact with the heating surfaces of the boiler. Heat is transmitted through these heating surfaces and turns the water, which the boiler contains, into steam. This heat is abstracted from the gases, which then pass away to the chimney.

Furnaces can be *internal* or *external*. With the former, combustion takes place in flues inside the boiler, and the fire grate or the furnace is surrounded or partly surrounded by water. With the latter, the process of combustion takes place outside the boiler proper, and the grates are placed underneath or sometimes in front of the boiler.

In all cases, but more especially with internally-fired boilers, some of the heat is imparted to the heating surface by radiation. The remainder of the heat reaching the water does so by the convection of the hot gases, and by conduction through the medium separating these gases from the water.

Evolution of the Modern Boiler.—To a greater extent than any other part of a steam plant, the generation of the steam itself has been developed empirically, so that present-day boilers are more the outcome of long practical experience than of theoretical considerations.

There are several natural reasons for this state of affairs. Steam was required for power purposes long before applied chemistry turned its particular attention to the composition of fuels and to scientific combustion; the principles of heat transmission, convection currents,

and radiation had not been stated in a form accessible to engineers; the manufacture of boilers was a cumbersome process; lastly, the average conditions under which boilers worked were so variable that it was instinctively felt that results obtained on a laboratory scale, with their finer shades of measurement and calculation, would be of little help in guiding practice.

Nevertheless the methods employed in the evolution of the modern boiler have proved far from unsatisfactory. Whereas the amount of heat appearing as useful work in the best steam engine is 20 per cent.[1] of the heat supplied to it, twenty years ago Bryan Donkin [2] gave the average results of ten experiments of Lancashire boilers as 72 per cent. thermal efficiency, and this is a type of boiler which owes its design to practical considerations. A modern water-tube boiler plant, including economisers and superheaters, is quoted by E. M. Lacey,[3] in his paper on the new power house at Birchills, Walsall, as showing an overall thermal efficiency of upwards of 88 per cent. on the lower or net calorific value of the coal. Such results correspond more to the mechanical efficiency of a prime mover and leave its thermal efficiency a long way behind.

Thermal Efficiency.—In view of the advances in our knowledge of the properties of steam, made in recent years, it is worth while to consider how the thermal efficiency of a boiler should now be measured.

Where steam is wanted quickly, or in fluctuating quantities, such as in transport or for rolling mills, the performance or evaporative power of a boiler is an important consideration. On the other hand, when economy in working is a principal object, such as in large power stations, or in most factories, the thermal efficiency of the boiler plant should take first place. This latter represents the ratio of the amount of heat transferred to the steam to the heat available in the fuel, and is usually stated as a percentage. Various ways have been used to arrive at the values of these two quantities; the method given here takes into consideration certain refinements which are now recognised to be correct, but which are not always considered necessary in practice.

The amount of heat contained in unit weight, or in the case of gas in unit volume, of fuel is called *the calorific value of the fuel.* It is measured in centigrade heat units (C.H.U.) or in British thermal units (B.Th.U.). These units are, approximately, the heat required to raise 1 lb. of water 1° Centigrade or 1° Fahrenheit, as the case may be. Since 1° F. is $\frac{5}{9}$ of 1° C.

$$1 \text{ C.H.U.} = 1\cdot8 \text{ B.Th.U.}$$

On the continent the kilo-calorie is used, but the calorific value of fuel in C.H.U. per lb. is the same as its value in kilo-calories per kilogram.

It may be here noted that Joule[4] found that one B.Th.U. was equivalent to 772 ft-lb. of work, and this value has been used for a large number of years. More accurate determinations, amongst

[1] W. E. Dalby, "Steam Power," 1915 ed. p. 263.
[2] "The Heat Efficiency of Steam Boilers," 1898 ed. p. 118.
[3] *Proc. Inst. Civ. Eng.* vol. cciv. 1917, p. 202. See also p. 43 of this book.
[4] *Phil. Trans.* 1850, also 1878.

others, by Rowland in America and by Griffiths and Callendar[1] in England, show that *the mechanical equivalent* should be 778 ft.-lb. The C.H.U. is then equivalent to $778 \times 1 \cdot 8 = 1400$ ft.-lb.

The only reliable way to obtain the calorific value of a fuel is by direct measurement in some form of calorimeter, but this gives the higher, or gross, value. That is to say, any hydrogen in the fuel on burning combines with oxygen to form water, and this is converted into steam by the heat of combustion and, in so doing, absorbs a certain number of heat units due to the latent heat of steam. In a boiler this steam escapes up the chimney with the flue gases as steam, but in the enclosed space of a calorimeter it is condensed again and gives back its latent heat to be measured. It is not fair to debit the boiler with heat which it could not receive, and so the thermal efficiency should always be calculated on *the lower, or net, calorific value of the fuel.*

This value is easily obtained if the percentage by weight of the hydrogen in the fuel is known. An example would make this clear. A certain Durham steam coal contains 4·6 per cent. of hydrogen, and has a gross calorific value of 14,350 B.Th.U. per lb. of fuel.[2] Since 1 lb. of hydrogen on burning forms 9 lb. of steam, $\frac{4 \cdot 6}{100} \times 9 = 0 \cdot 414$ lb. of steam is formed for every lb. of fuel burned. This is assumed to take place at atmospheric pressure, where the latent heat of steam is 970 B.Th.U. per lb. Hence $0 \cdot 414 \times 970 = 400$ B.Th.U. are unavailable, and the net calorific value of the fuel is

$$14{,}350 - 400 = 13{,}950 \text{ B.Th.U. per lb.}$$

Again, if the fuel is not dry when fired, a certain amount of heat is required to evaporate this moisture, and here again this will be lost to the boiler. Suppose that 1 lb. of fuel as fired contains 6 per cent. of moisture, and that 452 lb. are used per hour, of this 452 lb.

$$\frac{452 \times 6}{100} = 27 \cdot 1 \text{ lb. will be water.}$$

Hence the weight of dry fuel fired will be $452 - 27 \cdot 1 = 424 \cdot 9$ lb. per hour.

Supposing now 4050 lb. of feed water are used per hour, the weight of steam generated per lb. of dry fuel equals

$$\frac{4050}{424 \cdot 9} = 9 \cdot 3 \text{ lb.}$$

This figure represents the evaporative power of the boiler in question.

To find the amount of heat transferred to the steam it is necessary to know *the total heat* (H) *required to produce* 1 *lb. of steam.*

The water is first raised from the temperature of the feed to the boiling point corresponding to the pressure at which the steam is formed, it is then turned into steam at the same temperature and pressure, and thirdly (if required) it is superheated to a higher temperature at the same pressure.

[1] *Phil. Trans.* A. 1898, 1902, and 1912.

[2] Inchley, on "The Calorific Value of Solid and Liquid Fuels," *The Engineer*, vol. cxi. (1911), p. 155.

The first stage is often called the addition of sensible heat (h). For 1 lb. of water taken from freezing point to boiling point, this amounts to $212° - 32° = 180$ B.Th.U. at atmospheric pressure.

The second stage is called the absorption of latent heat of steam (L). This can be obtained by experiment or by calculation. 966 B.Th.U. at 212° F. for the latent heat is a figure used in many tables and empirical formulæ, such as Regnault's, but recent steam tables [1] agree that a more correct figure is 970 to the nearest whole number. This value is used in the present book.

In the third stage, if the steam is superheated at constant pressure, the extra heat added (h_s)

$$h_s = Cp_m(t_s - t)$$

where Cp = mean specific heat of the steam over the range of superheat at constant pressure

t_s = temperature of the superheated steam

t = temperature of dry saturated steam at the same pressure.

The value of the specific heat of steam used by Regnault [2] and others was 0·48, and this is the value recommended by the Institution of Civil Engineers in their standard form for tabulating the results of steam-boiler trials.[3] It is often taken as 0·5, but for some time now [4] it has been known that the actual specific heat of steam is not a constant quantity, and, whilst the above value is sufficiently correct at atmospheric pressure, for higher pressures its average value over the degrees of superheat should be used.

Fig. 1 gives this mean value for the degrees of superheat found in modern practice. It is based by his kind permission on Professor H. L. Callendar's steam tables (1915 edition), and should be read to two significant figures. This is quite accurate enough for boiler work. If greater accuracy is required the third significant figure may be calculated direct from the tables.

In this chart the temperature scale is the actual temperature of the steam, but the corresponding specific heat is the mean value between that temperature and saturation. Incidentally the curve of pressures gives the temperature of dry saturated steam from 50 lb. to 400 lb. per sq. in. absolute.

The total number of heat units (H) required per lb. of steam is the sum of these two quantities, sensible heat and latent heat (or three if the steam is superheated), that is

$$H = h + L[+ h_s] \text{ for dry steam.}$$

[1] The following are the values of L. at A.p. as given by modern steam tables :—

Mollier, 1906	970·4
Peabody, 1912	969·7
Marks and Davis, 1913	970·4
Smith and Warren, 1913	970·5
Callendar, 1915	970·7
Average value	970·34

[2] *Mem. de l'Acad. des Sciences*, vol. xxvi. (1862), p. 167.

[3] *Proc. Inst. Civ. Eng.* vol. cl. (1902), revised, vol. cxcv. (1913).

[4] Jakob und Knoblauch, *Zeitschrift V.D.I.* vol. li. (1907), p. 124. See also Knoblauch and H. Mollier, *V.D.I.* vol. lv. (1911), p. 665; and *V.D.I.* vol. lvii. (1915), p. 400.

Where a boiler, without a superheater, is being tested, it is commonly assumed that dry saturated steam is formed. In practice, steam is scarcely ever quite dry unless it is superheated. *Wet steam* is steam which contains a proportion of its weight as moisture

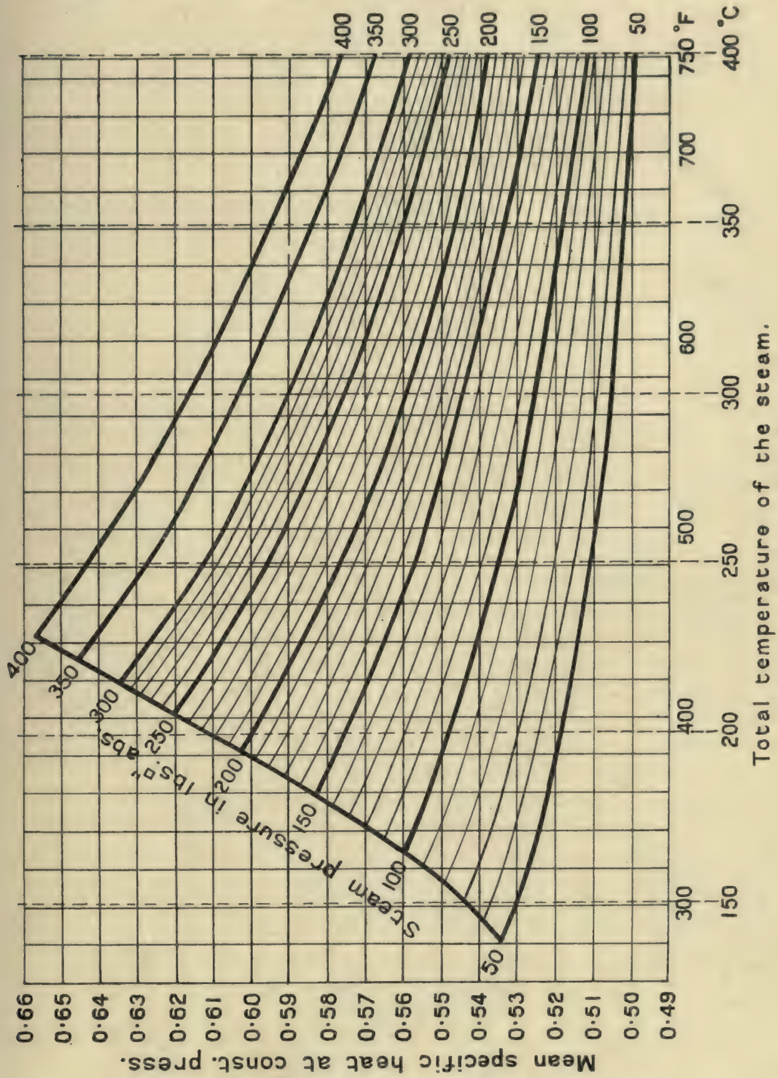


FIG. 1.—Mean specific heats of steam.

evenly distributed in it. Sometimes if a boiler is being forced and is too small to let the steam get away from the water surface, drops of water are carried away bodily, and the boiler is said to *prime.* Such water in mechanical suspension is often difficult to distinguish from wet saturated steam, which depends more on the lay out of the boiler and

the correctness of its design. When a boiler is working under normal conditions, however, the priming effect ought to be small, but if the steam still contains a small amount of water the boiler should not be credited with heat which it has not transferred to the steam. Suppose that the steam under these conditions has 1·5 per cent. of moisture, then the *dryness fraction* (x) of the steam equals

$$100 - 1{\cdot}5 = 98{\cdot}5 \text{ per cent.}$$

and the heat supplied by the boiler per lb. of steam

$$H = h + xL$$

The difficulty of obtaining a really fair sample of the steam as it leaves the boiler is very great,[1] and, as often as not, this factor is omitted. Once the sample is obtained, however, several satisfactory calorimeters [2] have been devised which give good results in determining the character of the steam.

The thermal efficiency (η) of a boiler is

$$\eta = \frac{\text{heat transferred to steam}}{\text{heat available in dry fuel}}$$

In the example quoted above

$$\eta = \frac{9{\cdot}3H}{13950}$$

since 9·3 lb. of steam were generated by 1 lb. of dry fuel.

Performance of Boilers.—The performance or evaporative power of a boiler is measured by the amount of steam generated per lb. of fuel. Sometimes the term is used as meaning the number of lb. of steam produced per sq. ft. of heating surface per hour.

In order to compare the performance of one boiler with another, it is convenient to adjust the actual number of lb. of steam produced per lb. of fuel to an *equivalent evaporation from and at* 212° F., that is, to the quantity of feed that could be turned *from* water at atmospheric boiling point into steam *at* atmospheric boiling point.

To obtain this a *factor of evaporation* is calculated, which equals

$$\frac{\text{total heat put into 1 lb. of steam}}{\text{latent heat of 1 lb. of steam at } 212^\circ \text{ F.}}$$

For wet steam, this is

$$\frac{(t - t_f) + xL}{970}$$

For dry saturated steam, this is

$$\frac{(t - t_f) + L}{970} \quad \text{or} \quad \frac{H + 32 - t_f}{970}$$

For superheated steam, this is

$$\frac{(t - t_f) + L + Cp_m(t_s - t)}{970} \quad \text{or} \quad \frac{H + 32 - t_f + Cp_m(t_s - t)}{970}$$

[1] See Report of the British Association, 1894, and Unwin on "The Dryness Fraction," *Proc. Inst. Mech. Eng.* 1895, p. 31.

[2] For description of calorimeters, see Royds, "Testing of Motive Power Engines," 1911 ed. pp. 185 *et seq.*; also Inchley, "Theory of Heat Engines," 1920 ed. p. 50.

where t = temperature of saturation in ° F.
t_f = temperature of feed water in ° F.
L = latent heat of the steam in B.Th.U. per lb.
H = total heat of the steam in B.Th.U. per lb.
Cp_m = mean specific heat of the superheated steam
t_s = temperature of the steam when superheated
t, L, and H are obtained from modern steam tables at the absolute pressure of the steam
t_f and t_s are measured direct
Cp_m can be read off Fig. 1.

For dry steam this factor does not differ more than 1 per cent. from the factor based on older steam tables, and often given in engineering pocket-books, but if the variable specific heat is taken into account (as it ought to be) it will affect the value for superheated steam in favour of the boiler.

To obtain the equivalent evaporation "from and at," the actual result is to be multiplied by the above factor.

If the equivalent evaporation per lb. of fuel, and the net calorific value of the fuel are known, the thermal efficiency of a boiler can be obtained as follows :—

$$\eta = \frac{\text{equivalent evaporation} \times 970}{\text{net calorific value of fuel}}$$

This is often useful when comparing boiler trials in which the efficiency is not stated. If the older value of 966 has been used in the trial it should be substituted for 970 in the above equation.

Boilers are rated as having a capacity of so many lb. of steam per hour. The term "horse-power," when applied to a boiler, usually meant an evaporation of 30 lb. of water per hour. The American Society of Mechanical Engineers[1] define one boiler horse-power as 34½ lb. of water evaporated per hour from and at 212° F.

Another rating, which is rapidly becoming obsolete, was nominal horse-power. This usually meant the amount of work that could be obtained from steam produced from 1 cub. ft., or 62·4 lb., of water.

It is customary to *measure as heating surface* all parts of a boiler that are in actual contact with flame or hot gases, and which, at the same time, have water or steam on the other side. The surface area is measured on the furnace side, though in America it used to be calculated from the outside diameter of all tubes, whether water tubes or smoke tubes.[2] In the American Code of 1915 this has now been altered to the furnace side as in Europe. American practice also distinguishes between surfaces below the mean water-level (water-heating surface) and any that may be above that level, which have steam on one side and products of combustion on the other (superheating surface).

If superheaters or economisers are installed, their heating surface and efficiency are recorded separately, though they may form an integral part of the boiler plant. This is to enable one boiler to be compared with another if different types of accessories are fitted.

[1] "Rules for Conducting Boiler Trials," *Am. Soc. Mech. Eng.* code of 1897.
[2] *Ibid.* It is, however, customary to measure the exterior of the firebox and smoke tubes in locomotive practice.

Definitions.—*Cylindrical boilers* may be described as those which contain all the water and steam in a vessel which is shaped like a cylinder. This cylinder may be placed horizontally in a brickwork setting, and is often then referred to as the *Drum type*, or it may be set upright on its own base, when it is known as a *Vertical boiler*.

If the furnace is placed in the brickwork the boiler is said to be *Externally fired*, but if it is placed in a firebox or in the flues, so that it is surrounded or partly surrounded by water, the boiler is said to be *Internally fired*. In classifying boilers it is better to use this distinction as a sub-division rather than as a main heading, because the fact of a boiler being fired internally or externally conveys little information as to its actual type.

The horizontal drum type of boiler may have one or more *flues* or large tubes running through it of sufficient diameter to take the fire grate in their front ends—such flues often have large water tubes across them, known as *Galloway tubes*, from the name of the firm which originated them. Sometimes a number of *smoke tubes*, or *fire tubes* as they are called in America, are included, which can be placed behind or above the flue tubes, or in the case of external firing which take the place of the flue tubes. Such boilers are called *Multitubular*. The function of these tubes is to increase the heating surface of the boiler (for a given grate area) by passing the hot gases through them on their way to the chimney.

If the furnace occupies the full length of the flue and a combustion chamber is arranged in the brickwork behind, the boiler is known as a *Dryback*. Such boilers are always multitubular, with the smoke tubes arranged inside the shell above, or at the side to return the hot gases to the front of the boiler.

In a brickwork setting the gases are usually returned again along the outsides of the boiler shell, which then count as heating surface. In marine work they pass up through a flue in front to the funnel.

In all these types the products of combustion are passed in tubes through the water, but it is possible to reverse the process and pass the water in tubes through the products of combustion. The latter form a very important class in modern practice, and are known as *Water-tube boilers*. They are often divided into *large-tube* boilers of 3 in. diameter or more, and *small-tube* boilers usually 2 in. or less in diameter. In nearly all cases one or more drums are required as reservoirs for the water and for the steam, and the tubes either run directly to these drums, or are connected to them through *headers*.

Except for vertical water-tube boilers which are internally fired, combustion nearly always takes place outside, and the path of the hot gases is controlled by suitable *baffle plates*.

When required, the steam after it is formed is brought into contact again with the hot gases to raise its temperature in *Superheaters*, and these often form an integral part of the plant. This is particularly the case with water-tube boilers.

Again, the water before it reaches the boiler is often passed through *Feed water heaters* or *Economisers*. The former use exhaust steam or other outside heat, whereas the latter use the flue gases after they have passed through the boiler. A third method is to augment the feed

with the exhaust steam itself after it has been reduced to hot water through a *Surface condenser.* This is a common marine practice where fresh water is limited, but is often employed on land as well, particularly in large installations.

The water is supplied to the boiler under pressure by *feed pumps* or *injectors*. The former require motive power to drive them, whilst the latter inject the steam which works them into the boiler along with the water. In this respect, therefore, they act as feed-water heaters.

The importance of using pure soft water for feed is becoming more and more recognised. Mechanically suspended dirt, such as mud or oil, can be removed by *feed-water filters* and *oil separators*, but chemically impure water should first be treated in a *water softening plant*.

The question of *fuel* is a large one, and can only be touched on here. Almost any substance—solid, liquid, or gaseous—that gives out heat, from sugar-cane refuse (bagasse) with a gross calorific value of 3500 B.Th.U., to petroleum with a gross calorific value of 20,000 B.Th.U., can be used in a suitably arranged combustion chamber.

With *solid fuels*, the chief of which is coal, the stoking is often done mechanically. *Mechanical stokers* fall broadly into three classes, depending on the way in which the fuel is fed into the furnace. Either the coal is spread evenly over the grate by *sprinkling stokers*, or it is placed on the dead plate in front and gradually worked through the furnace by *coking stokers*, or it is fed into the furnace from underneath by *underfeed stokers*.

Liquid fuels require no grate in themselves, but are sprayed into specially designed combustion chambers by *liquid fuel burners*.

Gaseous fuels fall broadly into two classes. Those in which gas is capable of being burnt in a gas-fired boiler, and those in which the *waste heat* is recovered by using it to generate steam in a waste-heat boiler. The principle of *surface combustion* is used in the former class. This is defined by Professor W. R. Bone, who discovered it, as combustion proceeding heterogeneously (as opposed to homogeneously), that is, in layers immediately in contact with an incandescent surface.

The waste heat from such sources as regenerative coke-ovens or reheating furnaces, puddling furnaces, or the exhaust of gas engines can also be partially recovered in an ordinary boiler, providing suitably large combustion chambers are used.[1]

Powdered fuels are now being considered and tried out, particularly in America.[2] Plants are required to pulverise and dry the coal, which is then delivered into specially designed combustion chambers by fans or blowers, mixed with air.

When the flue gases are drawn through the boiler by the action of a funnel or chimney, it is said to have a *Natural draught*. If mechanical means are employed, it is called either a *Forced draught* when steam

[1] See papers by T. M. Hunter, *Journal Inst. El. Eng.* vol. lvi. (1918) p. 57, and Arthur D. Platt, *Trans. Am. Soc. Mech. Eng.* vol. xxxviii. (1916), pp. 599–654.

[2] See "Report on Pulverised Coal Systems in America," by L. C. Harvey, 1919, published for the Fuel Research Board of the Department of Scientific and Industrial Research by H.M. Stationery Office.

or air are blown in under pressure, or an *Induced draught* if the products of combustion are drawn out by mechanical means.

In power-station work the modern tendency is to install both forced and induced draught. This has the great advantage of balancing the draught in the furnace and nullifying any tendency of leakage through the brickwork or other boiler casing.

Before proceeding to review the general question of boiler construction and design, it is advisable to consider, in detail, actual examples of modern boilers which show various types that are used in different countries. For the sake of reference these have all been placed together in the next chapter.

CHAPTER II

DESCRIPTIONS OF BOILERS

Boiler Types.—Examples described in this chapter have been selected as characteristic of modern boiler practice. In a book of this nature, which aims at taking a general survey of the various methods of obtaining power from heat, it is not possible to include the work of many excellent firms, whose designs are widely used, but which differ chiefly in detail from those shown here.

An attempt has been made to include only the latest designs in general use, and thanks to the courtesy of the many firms approached, it has been found possible to describe these in some detail. In many cases dimensioned drawings of a particular size are given, as very often more can be learnt about general design from the study of one working drawing than from the contemplation of a number of pictorial reproductions.

There appears to be no recognised method of classifying in detail the different designs of boilers. Nearly all authorities divide them into two classes, depending on whether the flue gases surround the water tubes, or are passed through the water in smoke tubes or flues, whilst a number lay stress on whether the boiler is fired externally or internally. The method adopted here is purely one of convenience, and embodies both these distinctions, the one as a sub-section of the other. On the other hand, the vertical principle is now used in so many different types that it is shown here as a variation where it occurs, instead of being classed as a single type by itself.

The following types are shown :—

CYLINDRICAL BOILERS

Externally fired . . .	American multitubular
	French elephant
Internally fired :	
Single flue	Cornish
	Cornish multitubular
	Vertical
	Vertical multitubular
Double flue . . .	Lancashire
	Lancashire with dished ends
	Yorkshire
	Galloway
	Dryback
	Scotch marine

CYLINDRICAL BOILERS—*continued*

Multiple flue . . .	Marine
	Bonecourt
Firebox	Locomotive

WATER-TUBE BOILERS

Externally fired:	
Large tube	Babcock and Wilcox land
	" " marine
	Niclausse land
	" marine
	Stirling land
	Woodison
	Nesdrum
Small tube	Yarrow
	Thornycroft
	Talbot
Internally fired	Hudson
	Vertical

American Horizontal Return Tubular Boiler (**H.R.T.**).—In Britain very few cylindrical boilers are externally fired, but in America the type of boiler shown in Fig. 2 is in common use.

It consists of a steel drum with a number of smoke tubes running through its entire length and expanded into the end plates. In its modern form the front end of the boiler is extended to form a smoke-box, as shown in this drawing. The boiler is placed in a brickwork setting, and suspended either by four lugs riveted to the shell plates and resting on wall plates, or by hangers from iron beams placed on top. Sometimes a three-point suspension is obtained by connecting two of these hangers to an equalising lever, which works on a central pin in the supporting beam.

The front end of the boiler is fixed, and the back end has slabs of firebrick attached to it, which take up the lateral movement of the boiler by sliding on the brick setting.

The surface of the end plates above the tubes requires strengthening as well as staying, and the tubes themselves are braced with longitudinal stays running through the boiler.

It is made chiefly in small or medium sizes, but is used, for all pressures, up to 500 American boiler horse-power (corresponding to an equivalent evaporation of 17,250 lb. per hour) and 5000 sq. ft. of heating surface.

Fig. 2 shows a boiler 7 ft. in diameter by 20 ft. long, which is rated at 300 H.P. for 185 lb. pressure. The heating surface is 3056 sq. ft. and the grate area 49 sq. ft.

The shell of this type of boiler is directly exposed to the fire, which may be a source of weakness. Special care has to be taken not to allow scale or greasy deposit to form inside. For the same reason the lower half of the circular seams are made as shown in the detail, to avoid a double thickness of plate.

The longitudinal seams require carefully designing, particularly for

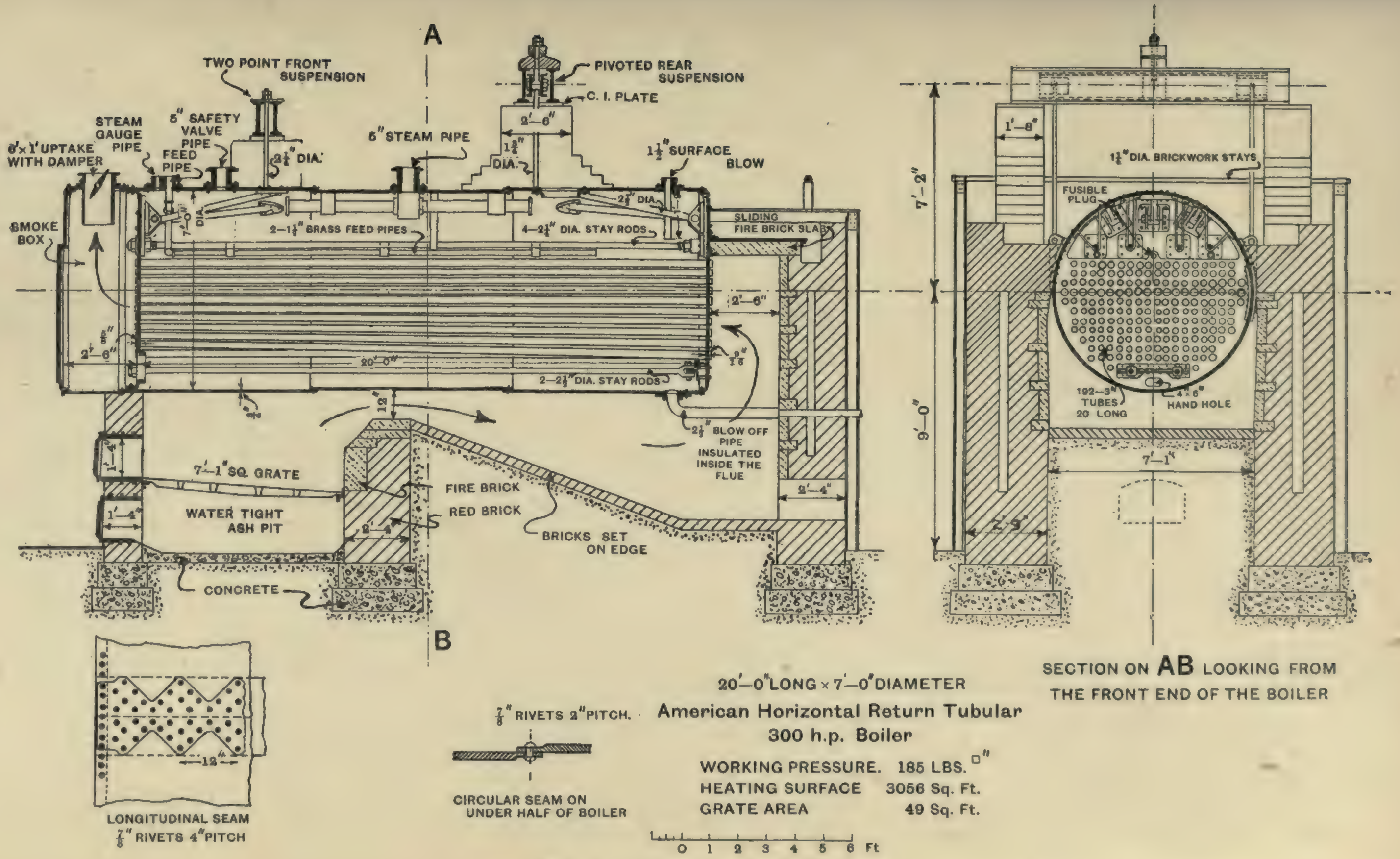


FIG. 2.—American H.R.T. boiler.

steam pressures over 150 lb. They should never be lap riveted, and cannot be made too strong. The saw-tooth joint shown was developed from German marine practice, and has four rows of rivets in a double cover-plate butt joint. The outside plate is cut away to obtain a very wide outside pitch without affecting the proper caulking of the joint. The plate efficiency of this joint is $\frac{12 - \frac{7}{8}}{12} = 92{\cdot}7$ per cent.

This design of boiler has a far greater heating surface and a considerably larger grate area than the internally-fired drum type. The ratio between the two is also much greater. In this case

$$\frac{\text{H.S.}}{\text{G.A.}} = \frac{3056}{49} = 62\tfrac{1}{4}$$

The larger grate area is more suitable for American fuel, and its low initial cost, compared with other cylindrical boilers of the same capacity, is probably the chief reason for its popularity in America.

The same boiler is also set on end in a vertical position, in which case it has a stayed firebox underneath. This would be about 8 ft. high for 20-ft. tubes. A smokebox is placed on top and the overall size of the shell would then be 30 ft. high by about 9 ft. in diameter. More details and tests of both the horizontal and vertical types of these boilers may be found in a communication by F. W. Dean to *Engineering*, vol. xcv. Feb. 7, 1913.[1]

French Elephant, Combined or Tischbein Boiler.—A boiler which is in favour on the continent and particularly in France,

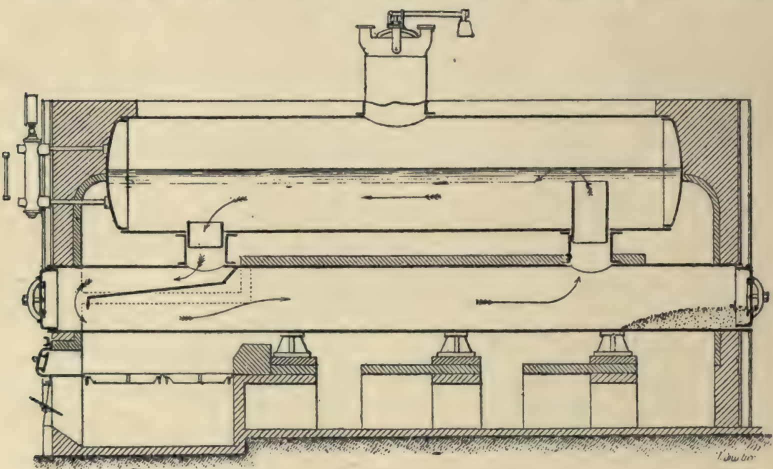


FIG. 3.—French elephant boiler.

is called in this country the elephant type. In America it is referred to as the combined or Tischbein (table-leg) boiler. It is usually externally fired, and consists of two or more cylindrical steel drums,

[1] From which Fig. 2 is derived by kind permission of the editor. See also papers by O. C. Woolson and F. W. Dean, in *Trans. Am. Soc. Mech. Eng.* vol. xix. p. 781, and vol. xxxvii. p. 619.

placed one above the other. Figs. 3–7 shows five forms, which are in common use abroad, the first three of which are externally fired.

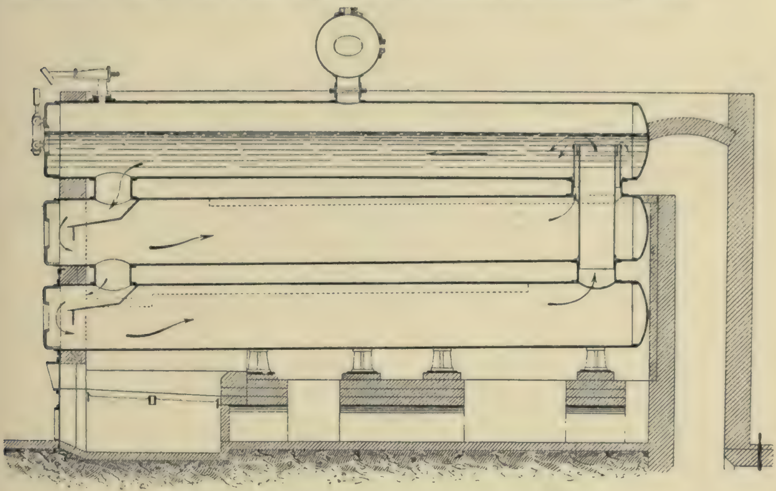


FIG. 4.—Multiple water-drum elephant boiler.

In its simplest form it consists of a cylindrical boiler with a water drum underneath (Fig. 3), which is known in France as a *Chaudière à bouilleurs*. The water drum is exposed to the direct action of the flames for a quarter or more of its length, and it supports the boiler

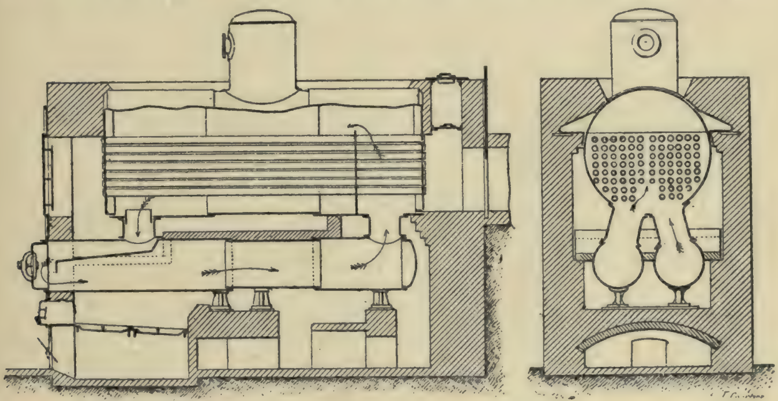


FIG. 5.—Semi-tubular elephant boiler.

proper on two short tubes of large diameter, through which the water flows. A baffle of firebrick between the drums causes the hot gases to flow the full length of the water drum before turning back along the underside of the boiler, whence they usually escape at the side or in front. When large supplies of steam are required, a number of water drums are placed one above the other as in Fig. 4. Two or three such

units are often built in side by side. Such boilers are referred to as *Chaudières multi-bouilleurs*.

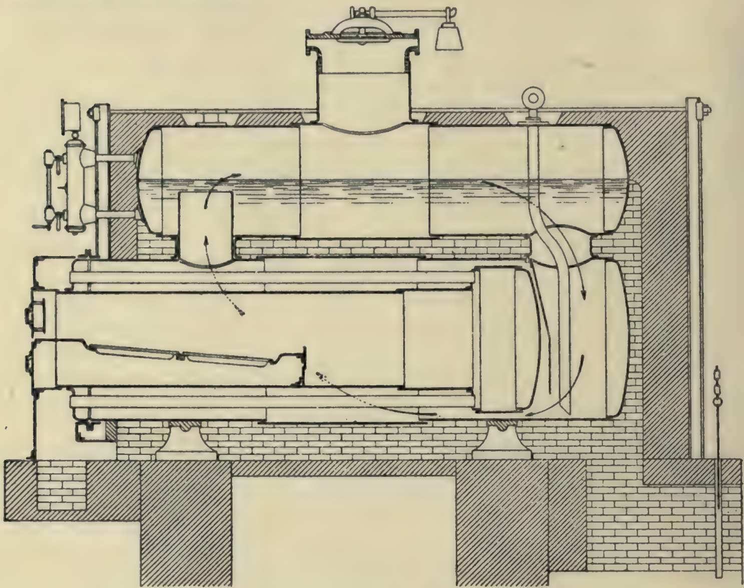


FIG. 6.—Internally-fired elephant boiler.

A more compact and efficient form is shown in Fig. 5 (*Chaudière semi-tubulaire*). Here two water drums, both in contact with the fire, are placed side by side, and the hot gases, after passing to the front of

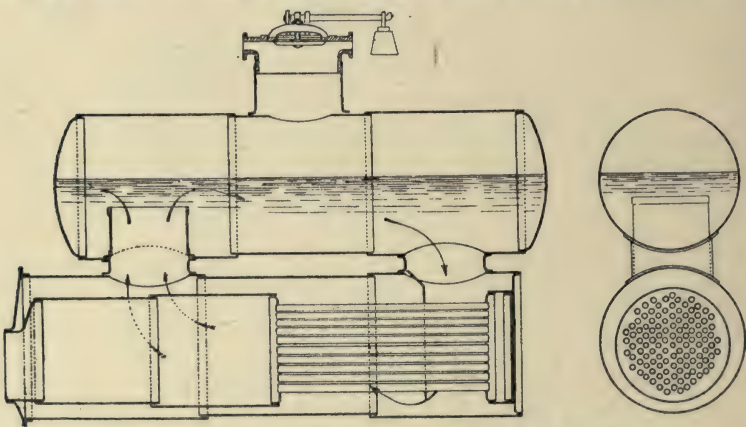


FIG. 7.—Semi-tubular internally-fired elephant boiler.

the boiler proper, are returned a third time through smoke tubes to the flue at the back end.

Figs. 6 and 7 are included to show the application of internal firing to this type of boiler, and are self-explanatory.

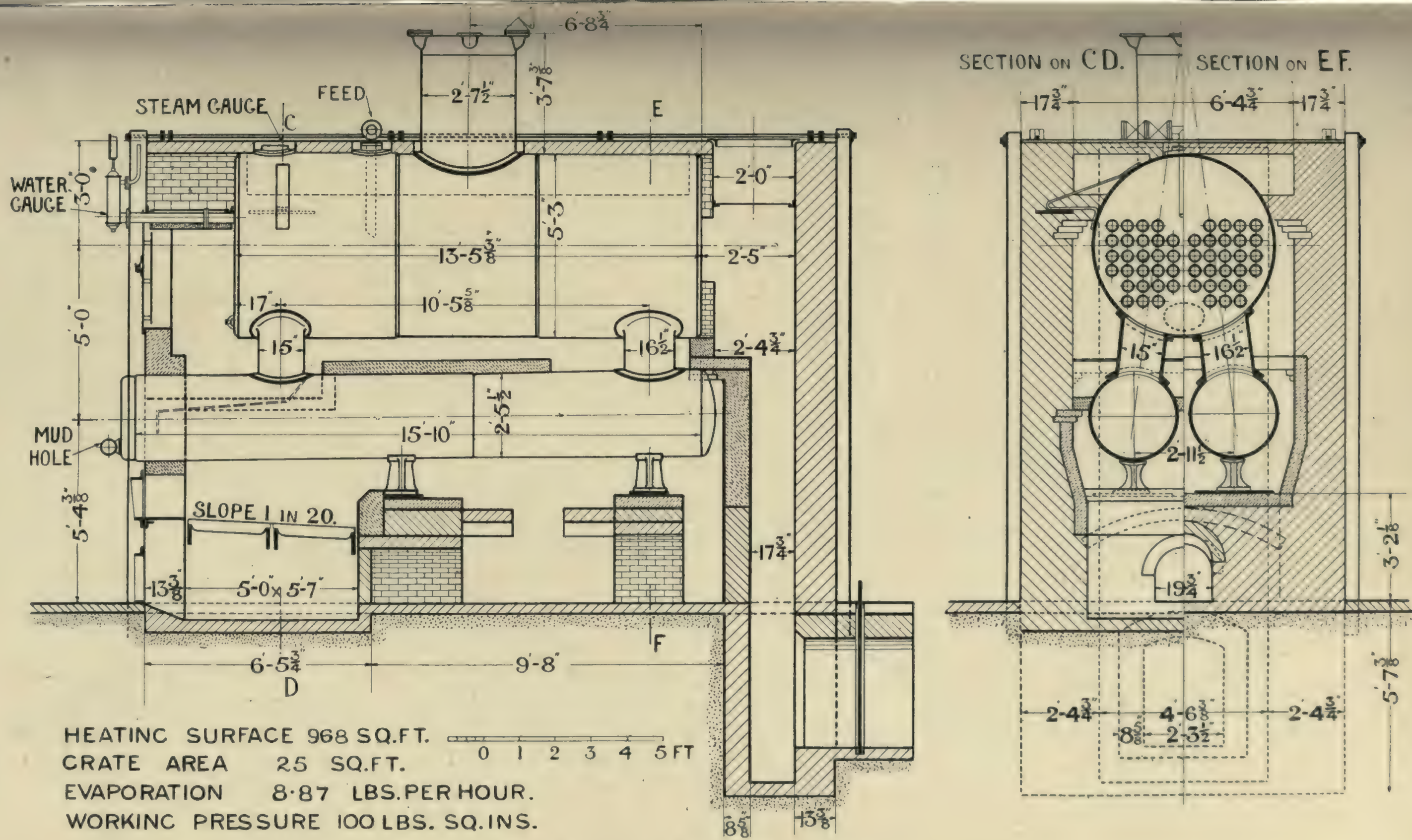


FIG. 8.—Modern French elephant boiler.

The steam dome, which is practically discarded in America, is largely used with the elephant boiler.

The semitubular type is made for duties from about 700 lb. of steam per hour up to 6600 lb. It has a reserve power of about 20 per cent. Fig. 8 shows one of the latest designs of this type. The weight of the upper drum is carried on the communication tubes at the back and on a pair of sliding brackets in front. A water-level indicator of the float type is fitted to the front flange on the top of the boiler, whilst the feed comes in vertically through the next flange. When the rear uptake is baffled, as shown in Fig. 5, the feed is brought in just in front of the baffle and discharged near the bottom of the drum. A lever safety valve is fitted on the top of the steam dome. The ratio of heating surface to grate area in this size of boiler is 38·8. To get the best out of anyparticular size of boiler, correct proportions of the brickwork setting are important.

The firm of J. Leroux and L. Gatinois, Ltd., of Paris, by whose courtesy Figs. 3–8 are reproduced, have made a special study of the control of water circulation in the elephant boiler. The A. Montupet system of baffles shown in these diagrams is a speciality of this firm. The steam which is generated immediately above the fire is prevented by means of a baffle, which is shown dotted in Fig. 8, from rising up the front communication tube, and has to sweep along the water drum, collecting more steam as it goes, to discharge itself up the rear header, which is extended nearly to the normal level of the water. This upward flow of steam and water keeps the whole circulation on the move in one direction, the water drums being also canted up slightly at the rear to assist the action.

Direct comparative tests on a boiler of the size shown in Fig. 8 give the evaporation per lb. of fuel (briquettes) as 8·6 lb. of steam per hour when fitted with these baffles as against 8 lb. without. An improvement of 7½ per cent., due entirely to controlled water circulation. The following trials, though not strictly comparative, show even

RESULTS OF EVAPORATIVE TRIALS OF A SEMITUBULAR ELEPHANT BOILER.

	Trial 1. Without baffles.	*Trial* 2. With baffles.	*Trial* 3. With baffles and enlarged grate.
Heating surface, sq. ft.	968	968	968
Grate area, sq. ft.	17	17	25
Ratio $\dfrac{\text{Heating surface}}{\text{Grate area}}$	57	57	38·7
Total water evaporated, lb.	19,699	23,434	31,854
Total coal burnt in lb.—			
Gross	3080	3157	4404
Net	2651	2717	3590
Evaporations—			
Per lb. of fuel, gross	6·39	7·42	7·23
,, ,, net	7·43	8·62	8·87
Per sq. ft. heating surface	2·07	2·43	3·29
Lb. of coal per sq. ft. grate area per hour	18·1	18·6	17·8

better results. The boiler, which is of the semitubular elephant type, was installed in the Lagny Depôt of the Paris General Omnibus Company. It was first tested as originally designed. Montupet baffles were then put in and it was tested again. A third test was made after the grate area had been increased. In each case the figures given here are the average of two trials of ten hours each, run on two consecutive days. A month intervened between each set of trials.

Internally-fired Cylindrical Boilers.—This class represents the main land types in common use in British engineering practice. In one sense the Scotch marine as well as the locomotive boilers are also included in this class, but they possess distinctive features both in

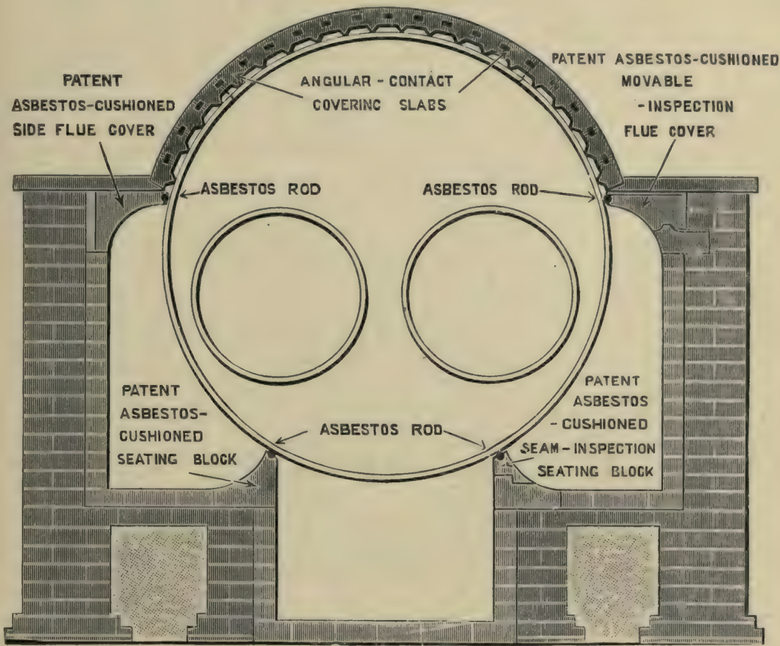


FIG. 9.—Lancashire boiler setting.

design and in performance, which usually cause them to be considered by themselves.

The internally-fired cylindrical boiler may be constructed with one, two, or more flues, according to the amount of heating surface required, and sometimes a number of smoke tubes are included as well. For small and medium powers the cylinder may be placed on its own base, when it occupies less floor space and does not require a brickwork setting. It is then known as the vertical boiler, and is always made with one flue or firebox, though sometimes smoke tubes are added to increase the heating surface.

The horizontal type can be used for the largest duties, but requires brickwork flues to lengthen the path of the hot gases and to prevent

undue radiation losses. A modern setting for the Lancashire type of boiler is shown in Fig. 9, reproduced by kind permission of E. J. and J. Pearson, Ltd., of Stourbridge. The danger of corrosion from moisture absorbed by the bricks is reduced by asbestos cushioning, and the use of interlocking arch blocks takes up the expansion of the boiler and prevents air leakage into the flues. Firebrick covering slabs with pockets of stationary air between them and the boiler plates provide a more efficient and more easily removable protection against radiation losses than the more common lagging of some non-absorbant material. As is usual with Lancashire boilers, the hot gases are first deflected downwards to pass underneath the boiler, and then are turned back from the front along each side to the chimney flues at the extreme rear.

Cornish Boiler.—The best known example of the *single flue* is the *Cornish boiler*, so called because it was introduced by Trevithick over 100 years ago for supplying steam to pumping engines in the Cornish mines. It is still largely used for installations where a steady low rate of evaporation is required with medium steam pressures. It is generally made from 4 ft. 6 in. to 7 ft. in diameter, whilst its length varies from three to four times the diameter. The flue is made as large as possible, having regard to a breathing space of from 6 in. to 9 in., which is bounded underneath by the shell and above by the gusset-stay angles. The flue runs parallel throughout its length, and is sometimes fitted with Galloway cross tubes. These not only strengthen it, but also help the water circulation. The furnace is placed in the fore part of this flue.

A modern Cornish boiler is shown in Fig. 10. It is designed for a working pressure of 100 lb. per sq. in. The shell would have a factor of safety of five, and be tested with water to a pressure of 180 lb. per sq. in. The circular seams are single-lap riveted, and the longitudinal joints double-cover plate with two rows of rivets. The front end is stayed with six gussets, and attached to the shell by a solid welded steel angle ring. The back end is flanged and also stayed with six gussets. All plate edges would be planed and fullered, all rivet holes drilled, and the riveting done by hydraulic machines.

The flue is made of soft mild steel formed into rings, and so arranged with Adamson caulking rings ($2\frac{1}{2}$ in. $\times$ $\frac{1}{2}$ in. thick) that no rivet heads are in contact with the fire or hot gases.

The fire grate has detachable C.I. grate bars 3 ft. long, and is backed with a firebrick bridge.

The feed pipe runs along the side of the boiler and is supported by a hook. It terminates at the front end plate in a wrought-steel feed block, to which a $1\frac{1}{2}$-in. feed check valve would be attached. The anti-priming pipe has a number of holes or slots running along the upper half of its circumference to ensure dry steam, and is connected to a 3-in. steam junction stop valve.

The manhole is made of the McNeill pattern, double riveted to the shell. This gives access to the interior of the boiler for cleaning and inspection purposes.

A pipe 4-in. in diameter carries a $2\frac{1}{2}$-in. dead-weight safety valve.

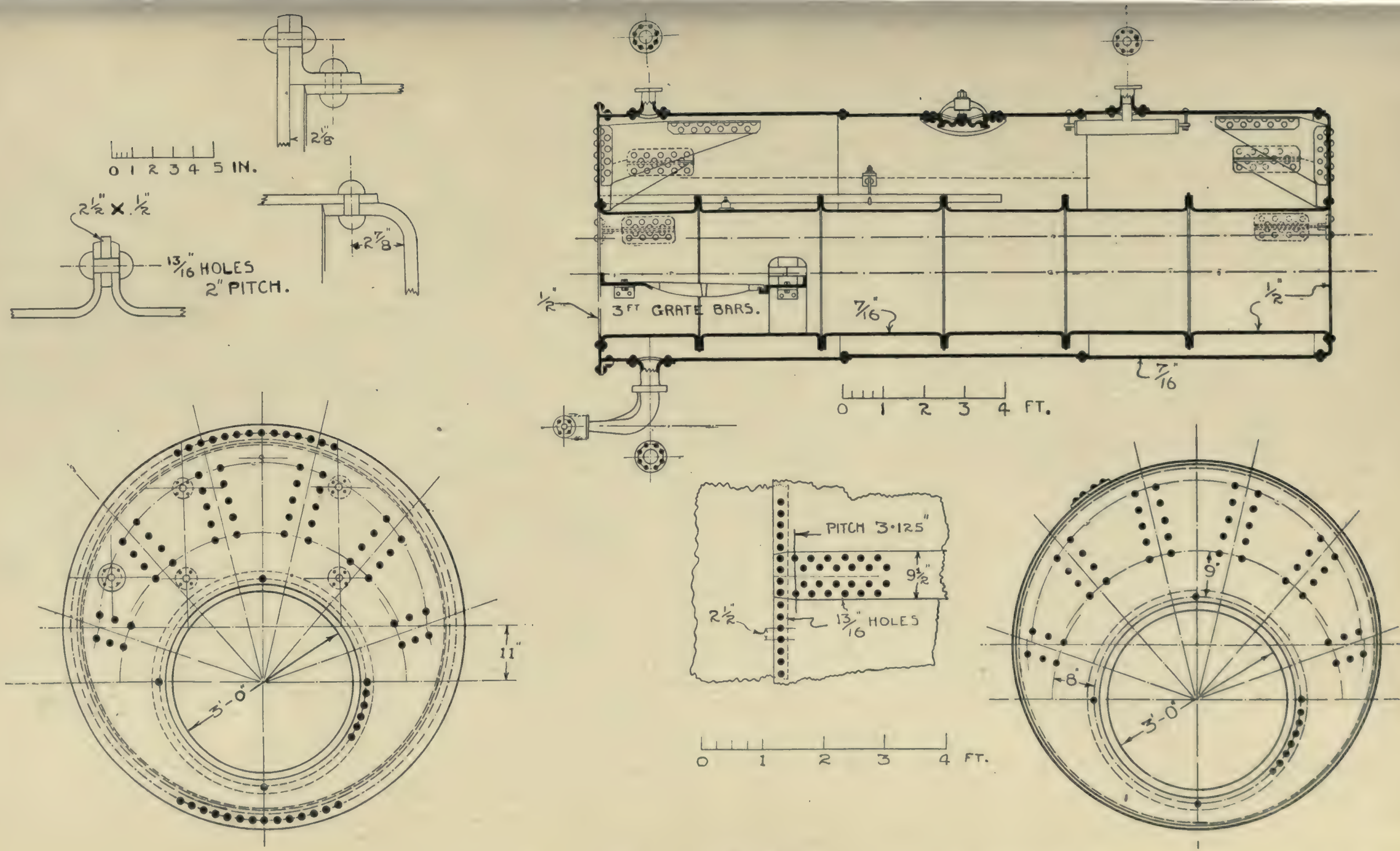


FIG. 10.—Cornish boiler.

A 3-in. blow off is fitted in front underneath with a cast steel elbow pipe, for which provision has to be made in the brickwork setting.

Such a boiler would be mounted in a setting similar to the one shown in Fig. 9 for a Lancashire boiler. There would, however, be no partition in the underneath flue. The hot gases after leaving the flue divide and pass along the sides of the boiler, and return underneath to the chimney at the back. In this way the hotter gases come in contact with the hotter parts of the steam and water, which is theoretically the best arrangement. It also has the advantage of lower temperatures underneath, should any scale be forming at the bottom of the boiler, with less likelihood of burning the shell plates. Occasionally the direction is reversed, and the flue gases pass along the bottom flue first, and finish along the sides; but this arrangement is more common when the boiler has two flues.

For shipment abroad, or when floor space is very limited, the *Cornish multitubular boiler* is economical and satisfactory. It should be used with soft water of good quality. The increased heating surface of the tubes makes the boiler smaller than ordinary one-flue or two-flue boilers of the same power.

Fig. 11 shows a Cornish multitubular boiler 16 ft. long by 6 ft. 6 in. in diameter. The general construction is practically the same as for the Cornish boiler just described. There are four $\frac{1}{2}$-in. thick gussets instead of six at each end, and four strong lifting lugs are fixed with $\frac{7}{8}$-in. studs, which are caulked round when the nuts are on.

There are ninety 3-in. plain tubes swelled $\frac{1}{16}$ in. at the back end, and eighteen 3-in. stay tubes swelled to $3\frac{1}{4}$ in. at the back end. These latter are screwed and fitted with two nuts, one inside and one outside the boiler plate.

A steel caulking ring joint is made between the front tube plate and the firebox. The fire bars in this case are 3 ft. 6 in. long. The fuse plug screwed into the crown of the firebox should be noted. This is arranged to melt before the crown plates get dangerously hot, and give ample warning to damp down or to draw the fire.

This type of boiler is sometimes made self-contained, the boiler sitting on cast-iron stools. In this case a steel smokebox and chimney are fitted to the back end of the boiler to carry off the flue gases. More usually the boiler is constructed to be built into brickwork flues similar to the Lancashire boiler setting in Fig. 9.

Figs. 10 and 11 have been taken from working drawings kindly supplied by Messrs. Holdsworth and Sons, Ltd., of Bradford.

Vertical Boilers.—For smaller powers and low pressures the single-flue cylindrical boiler may conveniently be set on end. It is then known as the *vertical boiler*, and the flue is usually called a *firebox.*

Fig. 12 shows a modern vertical boiler for 100 lb. per sq. in. working pressure, made by Messrs. Clayton, Sons & Co., Ltd., of Leeds. The firebox is surrounded as much as possible by water, and from two to four cross tubes are inserted to improve the circulation. In a good design these cross tubes are slightly inclined, to enable the steam to get away quickly in one direction as it is formed. The sides of the firebox are also slightly tapered for the same purpose. The

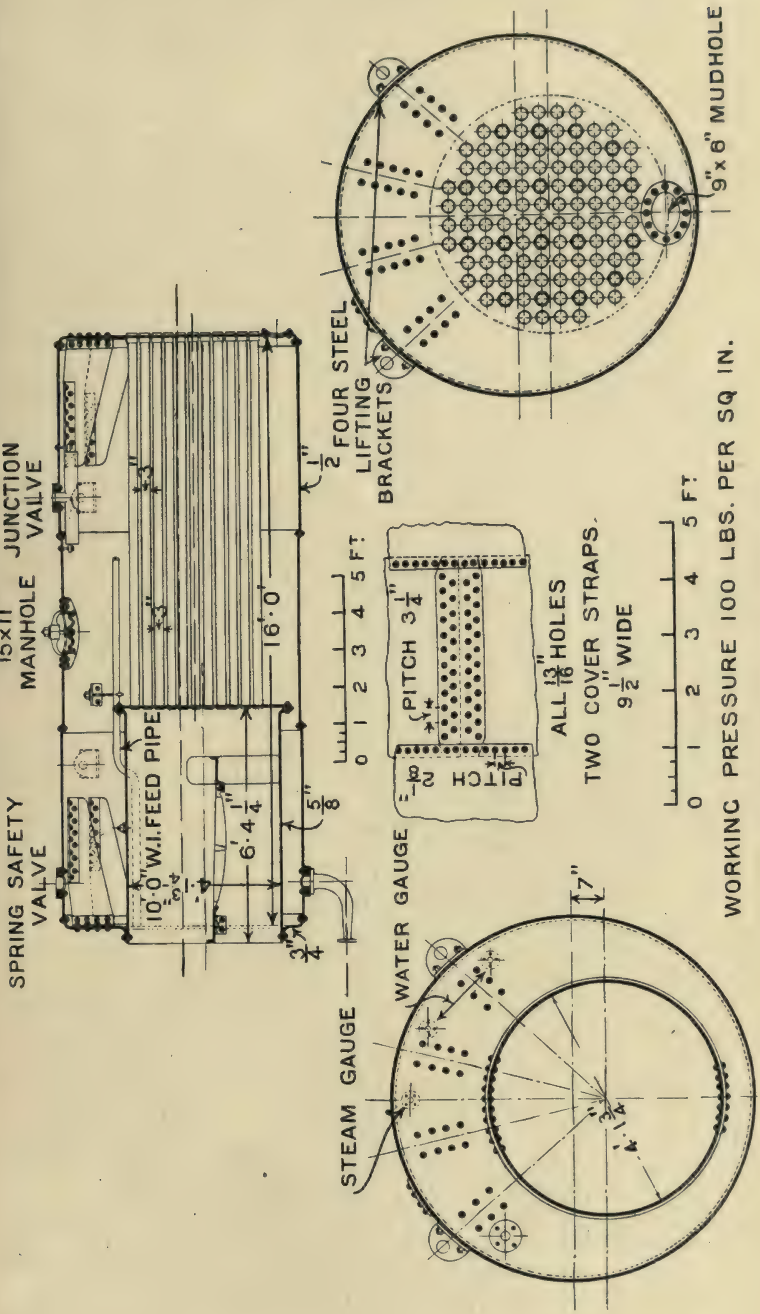


FIG. 11.—Cornish multitubular boiler.

firebox crown is cambered for flexibility, and the top of the boiler is usually dished for extra strength. The heating surface of this size is 190 sq. ft., whilst the grate area is $19\frac{1}{2}$ sq. ft. The capacity is 1000 lb. of steam per hour easy steaming.

The circular joints are single-lap riveted, and the vertical seams double-butt riveted. The firebox is double-lap riveted in larger sizes and single-lap riveted in smaller ones. A safety valve, steam-pressure

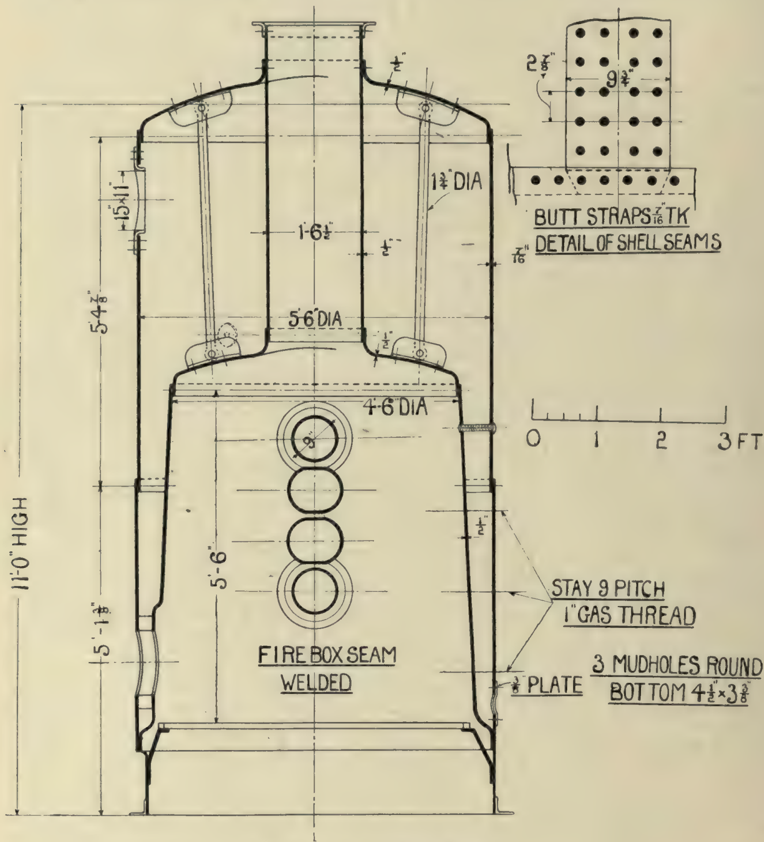


FIG. 12.—Vertical boiler.

gauge, and water-level indicator are always fitted, and a manhole provided for cleaning purposes and inspection.

Small vertical boilers, or donkey boilers as they are often called, are convenient for steam navvies, cranes, or other portable steam machinery, because they take up little room and require no brickwork setting. The surface of the water from which the steam is generated is small when compared with their grate area. They are therefore apt to prime and deliver wet steam when at all forced.

This small ratio of heating surface to grate area, usually about ten,

prevents the ordinary vertical boiler from reaching a high thermal efficiency. This is not a serious objection with small sizes, particularly if only used intermittently, as the total amount of fuel is small.

For steady steaming, especially in the larger sizes, it pays to increase the heating surface with smoke tubes. These tubes are sometimes arranged vertically above the firebox in the place of the single funnel of the ordinary type, but in such a position are only surrounded by water up to the free water-level. It is better to build the whole firebox on the skew and run the tubes horizontally, as is done in the Blake multitubular boiler, or to extend the firebox up one side, as is done by the Cochran Co., of Annan, in Scotland.

A normal *Cochran vertical multitubular boiler* for burning coal is shown in Fig. 13. The furnace, which has no riveted seams exposed to the flame, is pressed from a single plate by hydraulic machinery. The size shown is 6 ft. internal diameter by 14 ft. total height, and 21 of the 136 tubes are screwed at both ends to act as stays. The top of the combustion chamber is strengthened in the centre by a gusset, and the steam drum is hemispherical, which is the ideal shape for resisting internal pressure.

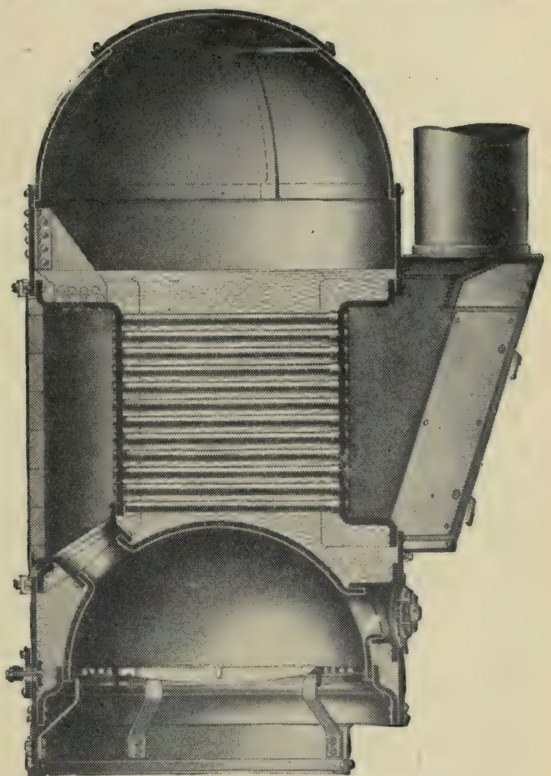


FIG. 13.—Cochran vertical smoke-tube boiler.

This boiler has a heating surface of 400 sq. ft. and a grate area of $18\frac{3}{4}$ sq. ft. The working pressure is 100 lb. per sq. in., but the type is made for pressures up to 150 lb. per sq. in. The evaporation is 2000 lb. per hour when burning coal at the rate of 260 lb. per hour, but the boiler can safely supply 2600 lb. per hour when forced.

The largest standard size evaporate 5000 lb. of steam per hour under easy steaming. This represents an evaporation of 8·2 lb. of steam per lb. of coal. It stands 17 ft. high, and is 8 ft. 6 in. in diameter, and the ratio of heating surface to grate area is 24·4. Such figures are every bit as good as the horizontal drum type of similar capacity.

Fig. 14 shows a Cochran boiler adapted for liquid fuel. The bottom plate is lined with firebrick, which is extended up the sides to protect the only riveted seam of the firebox. The hemispherical shape and the otherwise seamless crown of this firebox is especially suitable for withstanding the intense local heat which is present when burning oil fuel. The Cochran boiler, even for solid fuel, has a comparatively large furnace, and this, together with the brick-lined combustion chamber, provide the large volume which makes for efficiency when liquid fuels are used.

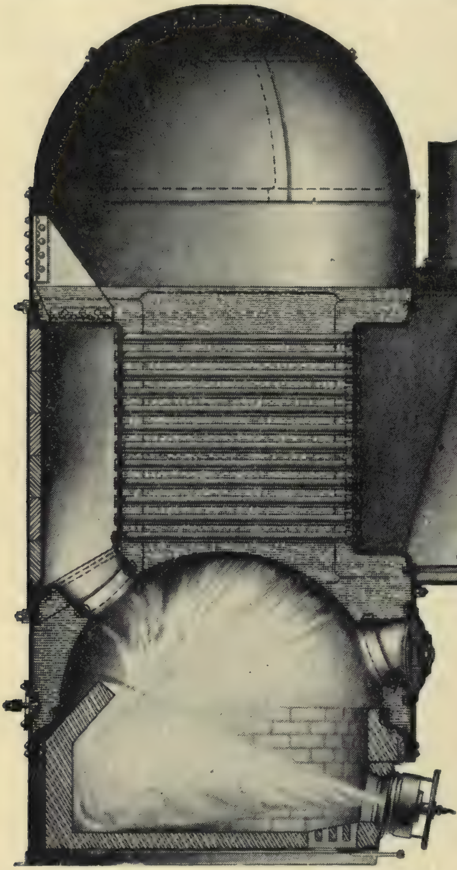


FIG. 14.—Oil-fired Cochran boiler.

A small air inlet with a sliding door is placed in the double-bottom plate for regulating the supply of air.

Another arrangement, where height is a consideration, allows for the oil to be fired through the fire door. This is sometimes convenient for donkey work or on board a ship.

The Cochran boiler may equally well be arranged to use waste gases from such sources as reheating furnaces, puddle furnaces, or hardening furnaces, where the temperature of the waste gases is not less than 1500° F. In order to ensure a large steam space and a large volume of water the normal type is increased some 2 ft. 6 in. in height. This enables steam to be withdrawn from the boiler in irregular quantities without unduly affecting the pressure, and provides an adequate thermal storage capacity to cope with a fluctuating heat supply.

The boiler should be mounted on a firebrick setting preferably immediately above the source of waste heat, and it has been found that the seamless furnace withstands temperatures of 2000° F., such as are sometimes found over puddling furnaces. The following table

shows actual normal evaporation obtained by this firm at a steam pressure of 100 lb. per sq. in. from feed water at 62° F. :—

Size of boiler.					Suitable for coal consumption in furnace of	Evaporation. Assuming waste gases reach boiler at undernoted temperatures.			Funnel.	
Dia.		Height.		Heating surface.		1500° F.	1800° F.	2000° F.	Dia.	Height.
ft.	in.	ft.	in.	sq. ft.	lb. per hour.	lb. per hr.	lb. per hr.	lb. per hr.	in.	ft.
6	0	16	6	400	260	800	1100	1300	26	60
6	6	17	0	500	320	980	1350	1600	29	60
7	0	17	6	600	390	1200	1650	1950	31	60
7	6	18	6	730	460	1420	1950	2300	36	60
8	0	19	0	850	550	1690	2320	2750	39	60
8	6	19	6	1000	600	1850	2540	3000	42	60

Lancashire Boiler—Much the commonest form of horizontal cylindrical internally-fired boiler in English practice is the *Lancashire boiler*, which has two parallel flues running the whole length of the drum. The design was directly developed by Sir W. Fairbairn from the Cornish boiler when larger capacities came into demand.

The general principles underlying its construction and operation are similar, but the boiler is made in larger sizes than is practicable with only one flue.

The proportions of the Lancashire boiler have been gradually evolved from long practice, and whilst most makers have their own range of standard sizes, the following table gives some indication of average dimensions in common use :—

Normal duty from and at 212° F.	Inside shell diameter.		Overall length.	Outside flue diameter.		Length of grate	
lb. of steam per hour.	ft.	in.	ft.	ft.	in.	ft.	in.
3500	6	0	22	2	3	4	6
4500	6	6	24	2	6	5	0
5750	7	0	28	2	7	5	6
7000	7	6	28	3	0	6	0
8000	8	0	30	3	3	6	0
9000	8	6	30	3	6	6	0
9750	9	0	30	3	9	6	0

A modern Lancashire boiler is shown in Fig. 15, by kind permission of Messrs. Joseph Adamson & Co., of Hyde, in Cheshire.

It will be noticed that the two flues are made as large in diameter as the outer shell will permit, and that they are tapered at the back end of the boiler. This is to allow a man to pass from the upper to the lower side of the flues for cleaning and inspection purposes. The back end plate is flanged, and the front end attached to the shell by a solid welded steel angle ring. Both ends are stayed with gussets, but a breathing space is left all round the flues. The furnace mouths are flanged outwards, and the flues themselves are built up in short lengths with Adamson caulking rings between each pair of flanges. In this way, not only is the flue well strenghtened to resist collapse, but no rivet heads are exposed to the fire or to the hot gases.

The front end of this boiler has provision for a steam gauge and for two water gauges, shown each side of the central gusset plate, and a mudhole at the bottom for cleaning purposes. Underneath is shown the flange to which is attached a blow-off elbow pipe, which is usually made of cast steel.

The check feed valve is often placed in front, though in this instance provision is shown for it on the top, leading to a 20-ft. distributing pipe, which extends just above the level of the flues along one side of the boiler. Flanges are also provided for two safety valves, the front one being a dead-weight safety valve and the rear one a high-steam and low-water safety valve. This valve blows off when the steam pressure rises above the working pressure, and also when the water falls below the ordinary level. The steam is drawn off through a junction valve, which connects to the C.I. anti-priming pipe inside the boiler. This pipe has both ends plugged up and a number of holes along the top half of its circumference only.

These boilers are always mounted in brickwork settings, of which Fig. 9 is a modern example. The hot gases are nearly always passed to the front underneath the boiler, and returned to the chimney flue at the back along the sides.

The Yorkshire Boiler.—In 1906 Mr. W. H. Casmay, of Wakefield, took out a patent[1] which embodied several interesting modifications of the Lancashire type of boiler.

The two flues, instead of contracting in area at the back end, are uniformly increased in diameter from front to rear, and at the same time are sloped upwards about 1 in 50.

This arrangement tends to reduce the density of the hot gases as they sweep along the flue. The resulting decrease in pressure induces a natural draught through the fires, which raises the furnace temperature and makes for more efficient combustion. The transmission of heat is greatest in front, where the larger amount of water is situated; but the increasing turbulence of the gases towards the rear should distribute the heat transmission more uniformly throughout the length of the flue than is the case with parallel flues with contracting ends.

It is found that for any given diameter the boiler need not be so long to obtain the same duty. This can be seen from the following table:—

Diameter.	Length in feet.		Normal duty from and at 212° F. lb. of steam per hour.	
	Yorkshire.	Lancashire.	Yorkshire.	Lancashire.
ft. in.				
6 0	17	22	3,500	3500
6 6	17	24	4,500	4500
7 0	20	28	6,000	5750
7 6	20 / 24	28	7,000 / 8,000	7000
8 0	20 / 24	30	8,150 / 9,400	8000
8 6	20 / 24	30	9,000 / 10,500	9000
9 0	24	30	11,400	9750

[1] British Patent, No. 22925 of 1906.

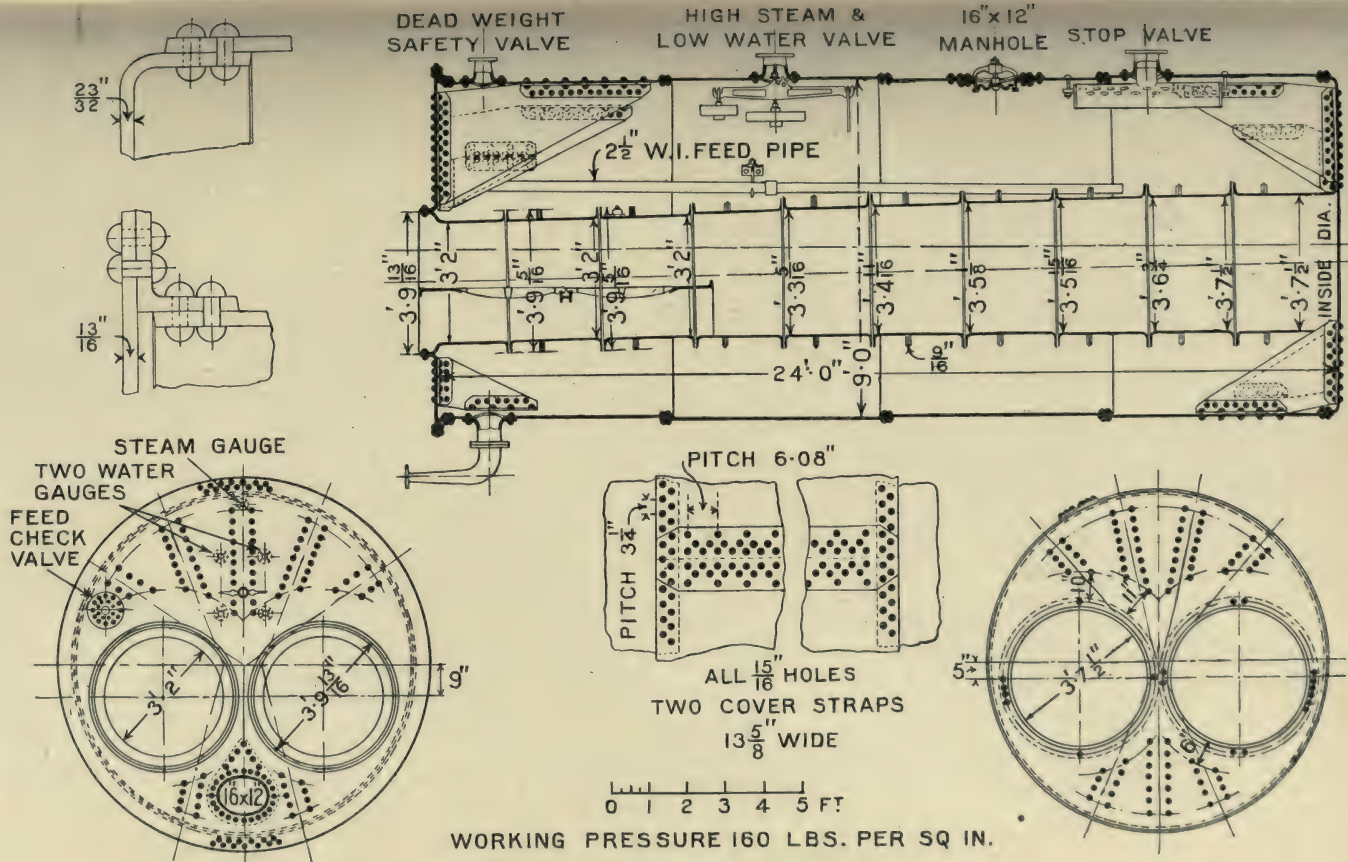


FIG. 16.—Yorkshire boiler (Holdsworth).

Another point embodied is a definite ratio of grate area to outlet area at the back end of the flue. In the Lancashire boiler this can vary between $3\frac{1}{2}$ and $2\frac{1}{2}$ to 1. In the Yorkshire boiler it is kept approximately at 1·8 to 1 for all sizes, which has been found a better proportion to ensure the proper escape of the hot gases.

The Yorkshire boiler is a speciality of the firm of Messrs. Holdsworth & Sons, Ltd., of Bradford, by whose courtesy Fig. 16 is reproduced. It will be noted that the flues at the front end are designed so that they may be withdrawn without dismantling the end plate.

The Galloway Boiler.—Another and an older modification of the two-flue Lancashire boiler is shown in Fig. 17, which is reproduced from drawings kindly supplied by Messrs. Galloways, Ltd., of Manchester. The two flues, each containing a furnace, are merged into one oval-shaped combustion chamber halfway through the boiler. This flue has a number of slightly conical water tubes passing across it, which are known as Galloway tubes. They serve the double purpose of staying the broad flat surface of the flue, and of increasing the heating surface exposed to the hot gases. By making the upper end of these tubes larger than the lower, any steam bubbles formed will get away quickly without undue obstruction, and tend to keep the water circulation in one direction. They were found to be so effective when first introduced that they are sometimes fitted to the flues of ordinary Lancashire or Cornish boilers. The Galloway boiler is made for all pressures and duties which are suitable for a Lancashire boiler, though the extra heating surface enables a shorter length of boiler to be used for the same duty. They are mounted in brickwork settings similar to Fig. 9, the flue gases passing to the front underneath the boiler and returning along the two sides to the chimney.

The Dryback Return Multitubular Boiler.—This type, sometimes known as the "Economic" boiler, consists of a cylindrical shell with one or two internal flues like a Cornish or Lancashire boiler, but the drum is made in much shorter lengths for a corresponding diameter. The water space above the flues contains a number of smoke tubes, and a combustion chamber is so arranged in the brickwork setting at the back that the hot gases, almost as soon as they leave the fire, are returned to the front of the boiler through these tubes. Here they either pass through a downcast flue direct to the chimney, or are again deflected to right and left round the outside of the boiler. The temperature at which the firebrick lining of the combustion chamber is maintained, owing to its proximity to the furnace, materially assists the combustion of the mixture of unburnt gases, and this fact, combined with the increased heating surface supplied by the smoke tubes, causes the final temperature of the flue gases to be from 150° F. to 200° F. lower than in the ordinary Lancashire type of boiler. A 14 ft. long by 8 ft. diameter boiler of this type has the same evaporative capacity from and at 212° F. as a 28 ft. long by 7 ft. 6 in. Lancashire boiler, namely, about 7000 lbs. of steam per hour. The boiler requires strengthening with longitudinal stays or plates in a similar way to the American H.R.T. boiler, and the tubes are made easily accessible, so that they can be kept clean by sweeping.

The Scotch Marine Boiler.—This type of boiler, which is very

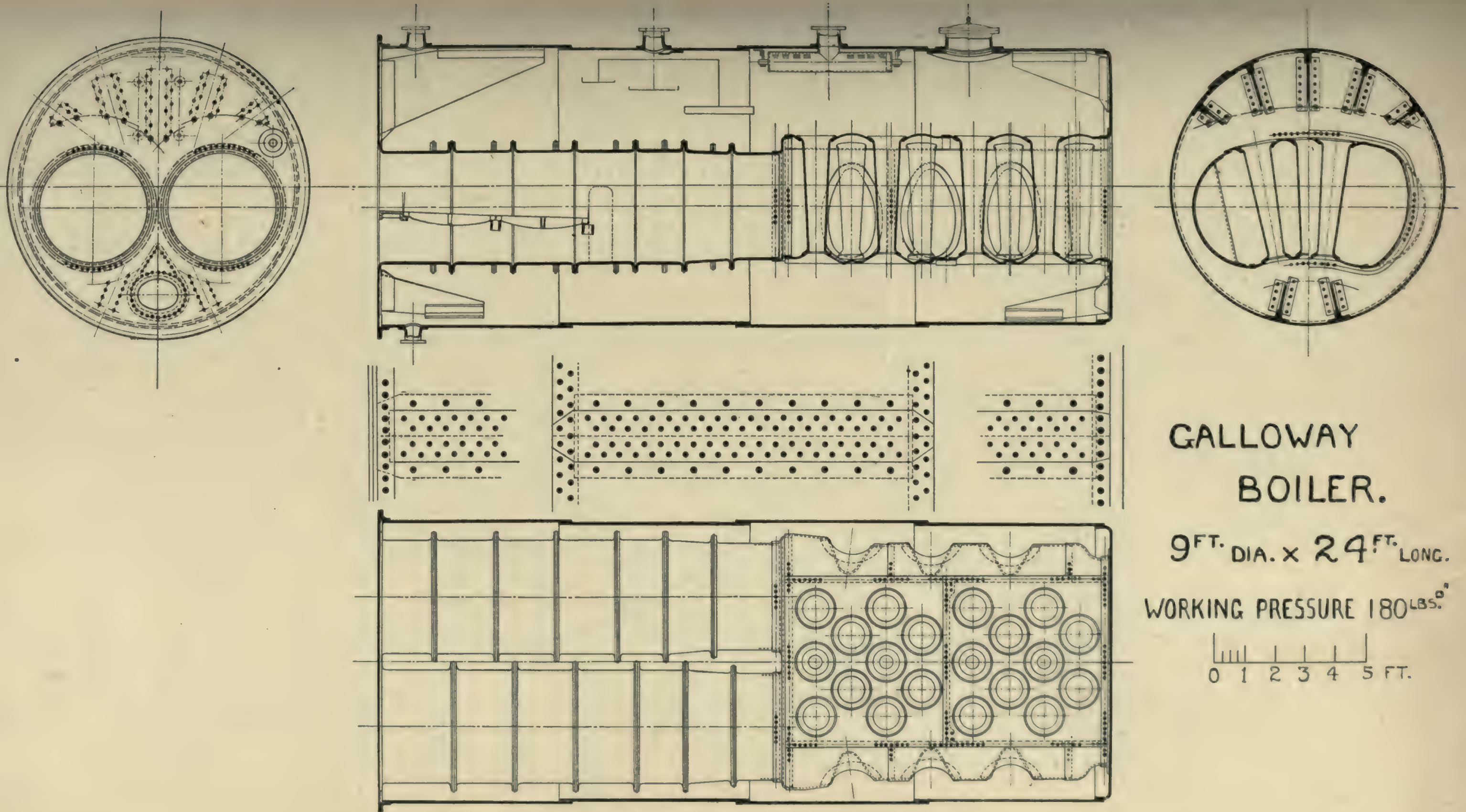


FIG. 17.—Galloway boiler.

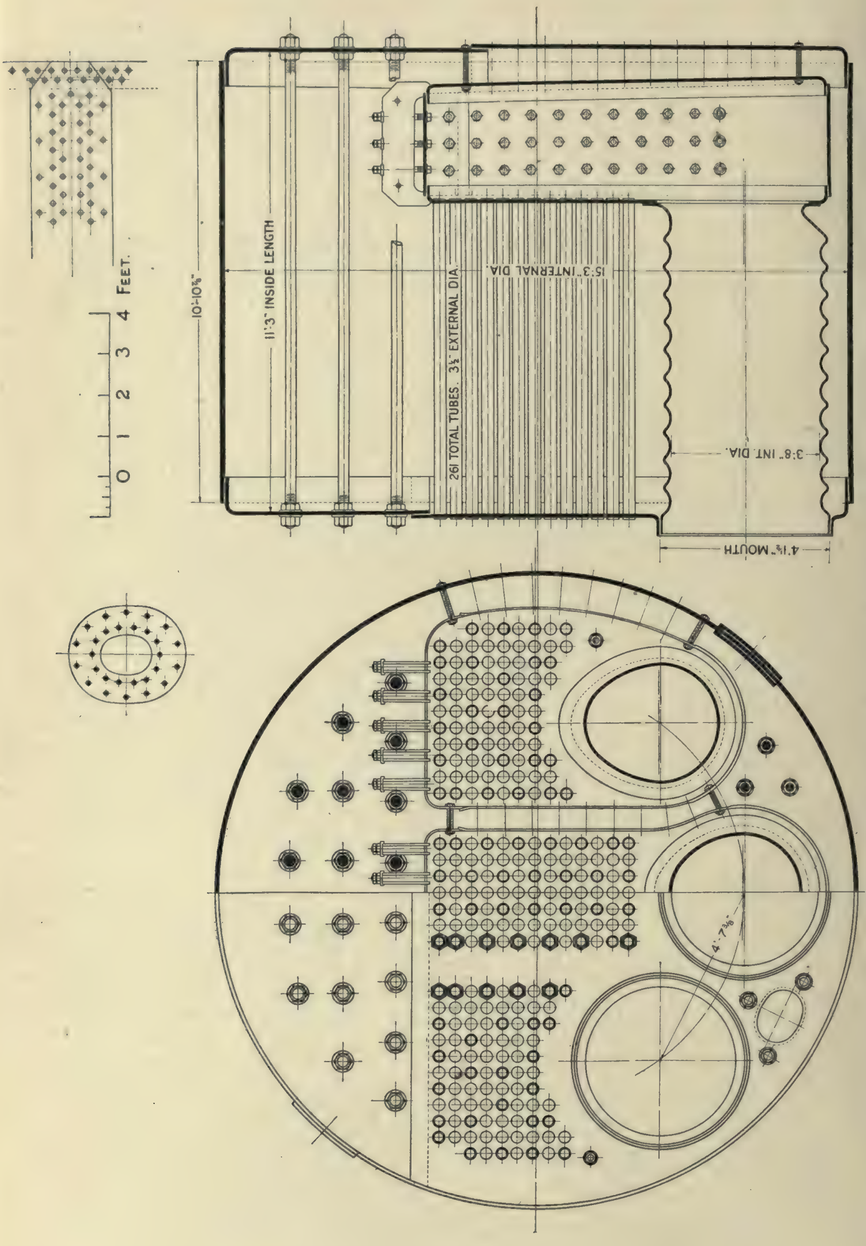


FIG. 18.—Scotch marine boiler.

common in British marine practice, was developed on the Clyde to provide a reliable steam raiser of large capacity in a restricted space. The diameter of the drum is increased and the length of the boiler curtailed for this purpose. Return smoke tubes are used, as in the Dryback type, but the combustion chamber is placed inside the water space and a separate one is made for each flue. A three-flue type of Scotch marine boiler is shown in Fig. 18.

This particular boiler, the drawing of which was kindly supplied by Messrs. Richardsons, Westgarth & Co., Ltd., of Hartlepool, embodies modern refinements which have been evolved by long experience. Corrugated flues of the Morison type are used with special egg-shaped ends, where they open out into their combustion chambers. These chambers are stayed to the outside shell and to one another, and are made of specially rolled plates, which are thicker underneath than at the sides. In this way the extra strain which comes on the lower part of this chamber is allowed for without riveted joints being necessary, and the only seams are made where the crown plate is inserted at the top. This flat crown requires staying by a number of girders as shown, and the front and back plates of the boiler are strengthened by longitudinal stays passing right through the boiler. The total number of tubes is 261, and their external diameter 2½ in. Owing to the variation of their thickness, it is customary to calculate the heating surface of these tubes on their outside diameter. Eighty-four of these tubes act as stays between the front plate and the combustion chamber, some being fitted with nuts and others screwed, 11 threads per in. The hot gases pass through these tubes, and are then carried away from the front of the boiler to the funnels. The area of the front tube plate, although in contact with these gases, is usually omitted when estimating the heating surface. In this boiler the heating surface is 2295 sq. ft. and the grate area 57·75 sq. ft., or a ratio of about 40 to 1.

The marine boiler is occasionally made double ended, with flues at both ends opening into common combustion chambers in the centre.

The Bonecourt Boiler.—The multiple-flue boiler has been developed to its logical conclusion in the *gas-fired Bonecourt boiler*. In this boiler the water is contained in a cylindrical drum, which may be placed horizontally or vertically, and any number of flue tubes, from five up to over a hundred, are run through from end to end. The inside diameter of these tubes varies from 2 in. to 9 in. in different designs, according to the fuel used, and they contain a refractory or iron spiral packing, which is raised to incandescence at one end when the burning gas passes over them. The packing itself is not consumed, but acts as a kind of catalytic agent, and is so shaped that the surfaces on which combustion takes place can freely radiate the heat generated on them to the water-cooled walls of the tubes. The radiation is so rapid in modern designs that the maximum temperature of the packing itself does not exceed about 1472° F. (800° C.). One form of refractory packing is shown in Fig. 19, and it will be noticed that ample room is left for any dust, which there may be in the gas, to be swept through the tubes without choking up the passage.

The principle of surface combustion, discovered by Professor W. A.

Bone, was originally applied to boilers by the late Mr. C. D. McCourt, and is now being developed by Mr. P. H. G. Kirke and the Bonecourt Waste Heat Boiler Co., Ltd., of London, who have kindly supplied the following drawings.

A modern gas-fired surface-combustion Bonecourt boiler complete with external superheater, economiser, and induced draught plant is shown in Fig. 20.

It consists of a horizontal drum 18 ft. long by 6 ft. diameter, through which 68 3-in. flue tubes are passed from end to end. Immediately behind the boiler is placed a superheater, containing a nest of curved weldless steel tubes placed horizontally across the path of the hot gases. The economiser behind the superheater also contains 68 3-in. tubes, and in both cases a central space is left so that a man can get between and reach all the tubes for cleaning and inspection purposes.

The gas issues from the two gas boxes in front through a number of jets corresponding to the tubes in the boiler shell. These jets are lighted, and drawn into the tubes together with air from the atmosphere by the induced draught of the fan shown behind the economiser. To get the mixture of air and gas correct when starting up, a pressure gauge is fixed on the gas box and a suction gauge on the fan. The

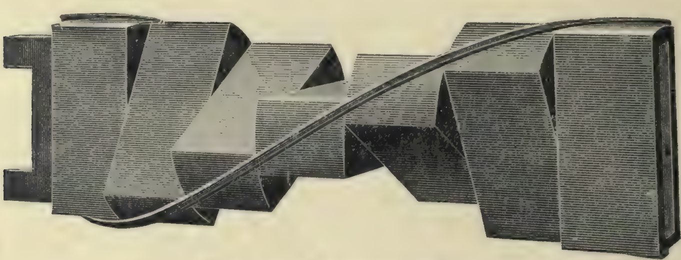


FIG. 19.—Bonecourt packing.

gas pressure is then regulated by the gas valve to suit whatever suction is on the fan, a table being given to the operator to guide him. The hot gases are arranged to leave the boiler shell at about 930° F., when they pass across the superheater coils and into the economiser tubes, which also contain iron packing. Here the gases are reduced to about 265° F., at which temperature they enter the fan without fear of corrosion.

Such a boiler requires no brickwork setting, and has a normal evaporation from and at 212° F. of 20,000 lb. of steam per hour when fired with coke-oven gas, that is twice the capacity of the largest two-flue boilers, or about three-quarters of that amount when fired with producer gas. It can be made to work at any steam pressure required in present-day practice.

The same principle can be used in boilers working on the waste heat of coke ovens or reheating furnaces, or on the exhaust from a gas engine. The temperature of the products of combustion available for the boiler varies from about 2000° F. in the former case to about 900° F. for the exhaust of a gas engine, but in either case the refractory

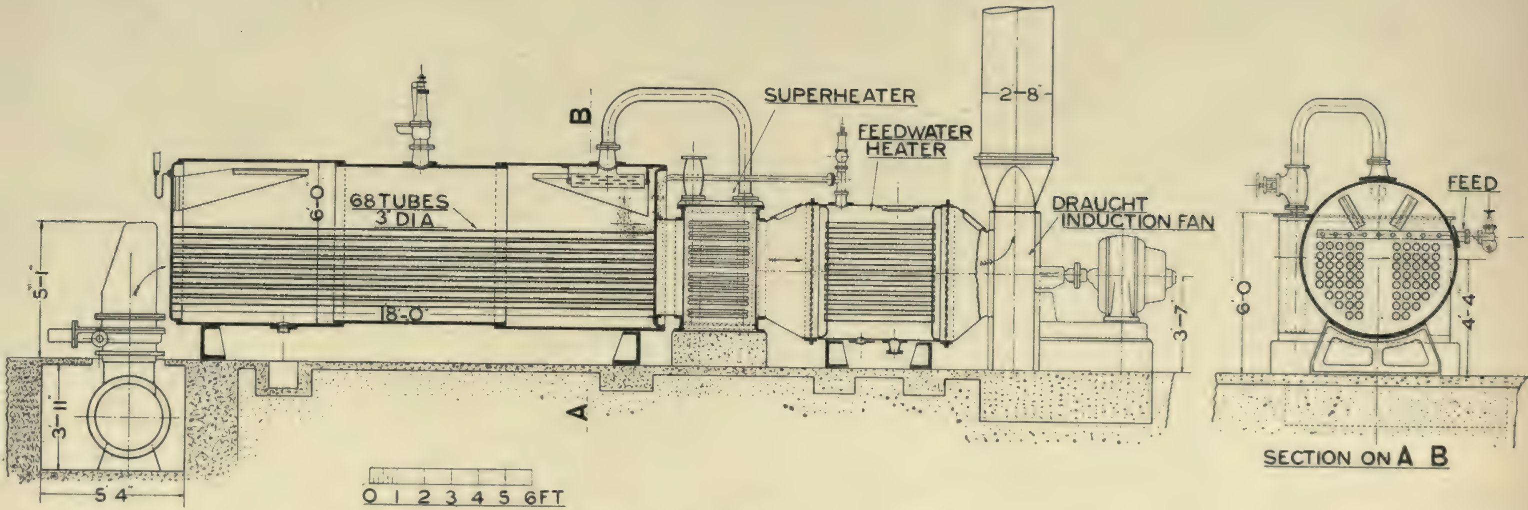


FIG. 20.—Bonecourt surface-combustion boiler.

packing and the boiler heating surface is so proportioned that the temperature on leaving the boiler is within 50° to 80° F. of the temperature of the steam, and no economiser is required. Except in the case of a gas-engine exhaust an induced draught fan is installed to create the necessary suction in the boiler. The drums themselves are similar in design to the one just described, and may be placed horizontally or vertically. No fan is required, and the packing deadens the sound waves to such an extent that the boiler takes the place of the silencer, which would otherwise be fitted.

Tests on Bonecourt Boilers

Boiler A.—Installed at the works of the Skinningrove Iron Co., Ltd., and tested by Mr. J. M. Whitham, in July, 1912.
Boiler B.—Installed at the London Works of the Bonecourt Boiler Co., Ltd.
Type.—Surface combustion with feed-water heaters and fan suction.
Flue tubes.—(Effective heating area)—
Boiler A.—109 tubes, 3 ft. 8 in. long and 3 in. internal diam.
Boiler B.—5 tubes, 10 ft. 6 in. long and 9 in. internal diam.
Total heating surface (of boiler only)—

Boiler A. $$109 \times 3{\cdot}66 \times \frac{3\pi}{12} = 314 \text{ sq. ft.}$$

Boiler B. $$5 \times 10{\cdot}5 \times \frac{9\pi}{12} = 123{\cdot}7 \text{ sq. ft.}$$

OBSERVATIONS AND DEDUCTIONS.

	Boiler A.		*Boiler B.*
	Light Duty.	*Heavy Duty.*	*Normal Duty.*
Duration of trial	5 hrs.	7 hrs. 2 min.	3 hrs.
Fuel—			
Description	Coke-oven gas		Oil
Cal. val. (B.Th.U.)	521 (per cub. ft. of gas at N.T.P.)	516	17,800 (net) (per lb. of oil)
Total per hour	8032 (cub. ft. at N.T.P.)	9920	181·7 (lb.)
Fuel per sq. ft. of heating surface per hour	25·4 (cub. ft. at N.T.P.)	31·4	1·47 (lb.)
Total heat available in fuel per hour (B.Th.U.)	4,184,880	5,118,720	3,234,060
Suction in duct connecting boiler and feed-water heater (in in. water gauge)	10·5	17·0	—
Do., connecting feed-water heater and fan (in in. water gauge)	11·3	20·3	—
Water evaporated—			
Total per hour (lb.)	3374	4245	2542
Steam pressure (lb. per sq. in. abs.)	112	112	125
Temperature of water entering feed-water heater (° F.)	64	61	43
Ditto, entering boiler (° F.)	131	128	119

	Boiler A.		Boiler B.
	Light Duty	*Heavy Duty.*	*Normal Duty.*
Water evaporated—contd.			
Dryness fraction per cent.	99·3	99·3	Not recorded
Factor of evaporation (including feed-water heater)	1·181	1·185	1·211
Equivalent evaporation, from and at 212° F. (lb. per hour)	3983	5030	3078
Ditto, per lb. of oil per hour	—	—	16·9
Ditto, per sq. ft. heating surface per hour	12·7	15·9	24·8
Gross thermal efficiency of boiler and feed-water heater	92·4	95·3	92·3
Power taken by fan (K.W. per hour	5·49	6·097	Not recorded

The Locomotive Boiler.—To meet the special requirements of the steam locomotive, a distinct type of boiler has been evolved, in which thermal efficiency is sacrificed for the rapid generation of a fluctuating supply of steam. It is hardly possible to do justice to the work and experience involved in the development of the locomotive boiler without taking up too much space for a book of this nature, more especially as the design of this type is intimately connected with the design of the whole locomotive. An exhaustive treatment of the subject appeared in serial form in *The Railway Engineer*, under the title, "Modern Locomotive Engine Design and Construction," by "M.I.Mech.E. and J. T. H."[1]

The principle upon which the type is based can be seen from Fig. 21, which is a diagram of a locomotive boiler arranged to burn oil fuel, by Messrs. John I. Thornycroft & Co., Ltd. Briefly, it consists of a firebox surrounded on all sides by water, and a cylindrical shell through which a large number of small-bore smoke tubes are passed to a smokebox at the front end. Superheaters of the Schmidt type are sometimes inserted in these smoke tubes from the front end, and a steam dome is usual from which to collect dry steam. The whole boiler requires careful designing, as it is subjected to very considerable strains.

EXTERNALLY-FIRED WATER-TUBE BOILERS (LARGE TUBES)

The main designs of this class of boiler fall into two divisions, depending on whether the water tubes are more nearly inclined to the horizontal, or to the vertical.

The boilers made by the firms of Babcock & Wilcox, Ltd., of Renfrew, the Babcock & Wilcox Co., of New York, and by J. & A. Niclausse, of Paris, are the chief examples of the horizontally inclined water tube. In the other division, where the tubes run more or less vertically, there are several well-known designs. Amongst others, the "Stirling," made by the Stirling Boiler Co.,

[1] London: 15, Farringdon Avenue, E.C. Commencing vol. xxxvi (1915).

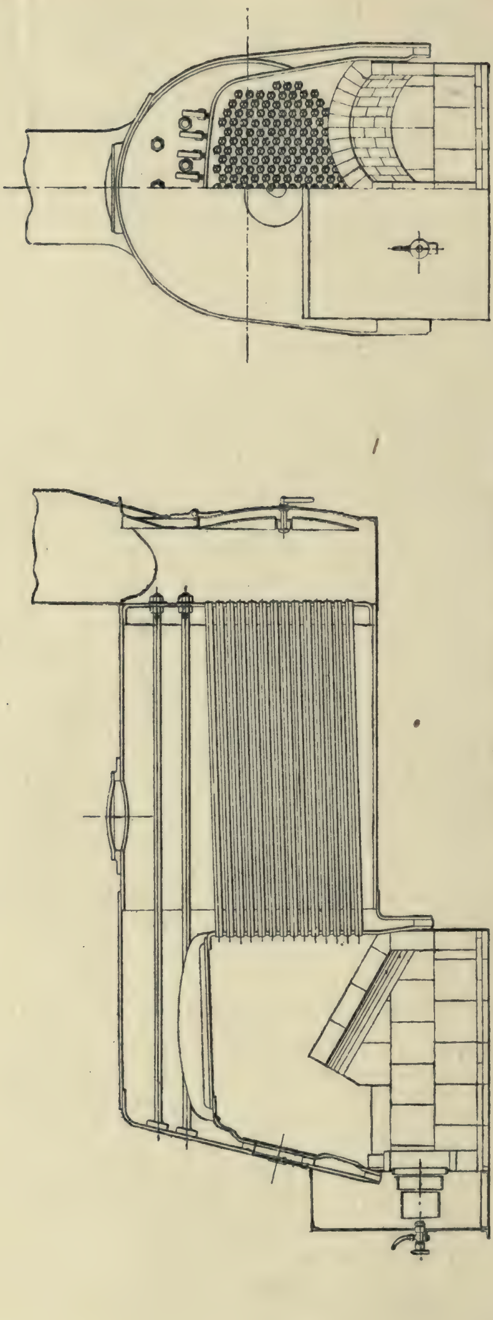


FIG. 21.—Locomotive type of boiler, oil-fired.

Ltd., of London and New York; the "Nesdrum," made by Messrs. Richardsons, Westgarth & Co., Ltd., of Middlesbrough; and the

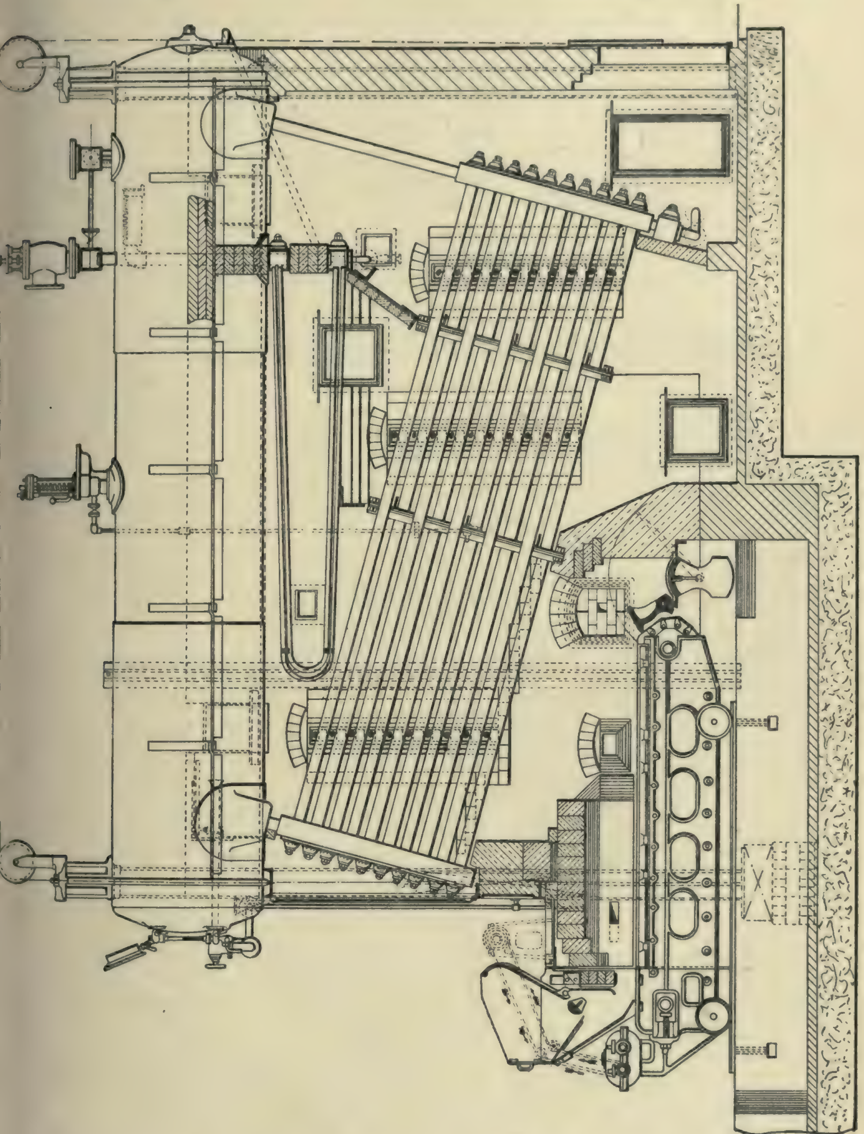


FIG. 22.—Babcock boiler.

"Woodeson" and the "Thompson," made by Messrs. Clarke, Chapman & Co., Ltd., Gateshead-on-Tyne, and by Messrs. John Thompson Water-Tube Boilers, Ltd., Wolverhampton, respectively.

The Babcock and Wilcox Boiler.—Present-day boilers of

this make embody the accumulated practical experience of over fifty years, a valuable asset in boiler design. The latest normal land type, with superheater and chain grate stoker, is shown in Fig. 22. It consists of a steam and water drum placed above, and in the same plane as, a "nest" of straight large-bore water tubes. The ends of these tubes open out into headers, which in turn are connected by tubes to the front and rear of the drum. The headers are made in one piece for each vertical row of tubes, and the tubes are so arranged that each horizontal row comes over the spaces in the previous row.

The tubes are inclined at about 15° to the horizontal, and have two baffles across them to control the paths of the hot gases. A second

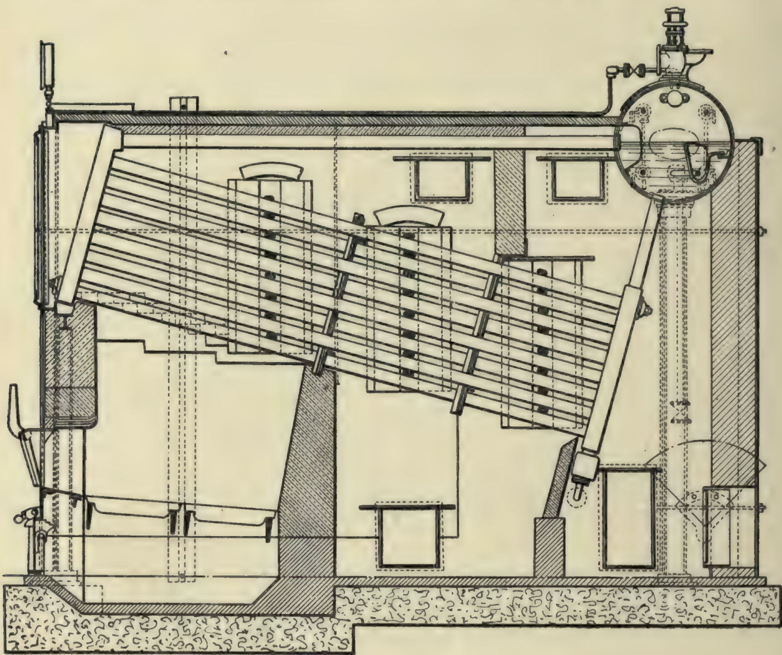


FIG. 23.—Cross-type Babcock boiler.

drum of square sectional area runs along underneath the back or lower headers, to which it is connected by a row of short tube lengths, and provision is made to drain off the water from the boiler at this point when required.

The superheater consists of two rows of expanding seamless steel tubes, connected at both ends with manifolds, the upper one receiving the saturated steam from the boiler, and the lower one delivering the superheated steam to the stop-valve through two pipes running up each side of the drum. A pipe connection from the lower manifold to below the water-level in the drum enables the superheater to be flooded, and to act as additional heating surface when the boiler is being started up, or when superheated steam is not required.

The steam and water drum is fitted with dished ends. A water-level gauge and a steam gauge are shown on the front end. The feed comes in at this end through a horizontal bell-mouthed pipe which carries just past the point where the front headers discharge into the drum.

A spring safety valve is mounted in the centre, and the stop valve has a bye-pass directly connected with the boiler.

The whole boiler is carried on slings at each end, and surrounded with brickwork, which has inspection doors in front, to allow free access to the hand-holes fitted to the ends of all the water tubes.

The hot gases impinge on the water tubes at right angles, which is a special feature of this boiler. After leaving the fire they cross about 50 per cent. of the length of the tubes. They are then deflected along the superheater by the drum and its setting, and are passed across the tubes twice more by means of the second baffle and the rear headers before escaping at the exit shown below at the back. The damper over this flue exit is controlled from the front by means of a chain passing over two pulleys. Manholes for cleaning purposes are provided in the brickwork, and a peep-hole is placed just above the grate for inspecting the thickness or condition of the fire at any time.

The water circulation is free, and a baffle in the drum above the front headers prevents the discharge of steam from breaking surface too violently.

Figs. 23 and 24 show the Babcock & Wilcox boiler with the steam and water drum arranged across instead of parallel to the water tubes. This "cross type," as it is called, is more convenient when space is limited or when difficult transport is a consideration. The drum, being shorter, provides less steam and water capacity, but otherwise this type has been found as satisfactory as the original design. Fig. 23 shows the cross type arranged for coal, hand-fired, and the brickwork setting is replaced by a steel casing lined with refractory material. Fig. 24 shows the portable type as arranged for wood firing. The grate area has been enlarged, and the two baffles controlling the path of the gases do not extend right across the tubes. Additional baffles laid on top and underneath ensure that the hot gases completely traverse the water tubes before reaching the flue exit above.

The cross type is largely used in marine work, where it lends itself readily to oil firing, but it is now being adopted on land for electric power stations. One of the reasons for this is the increasing necessity of economy in boiler house space to reduce capital costs. Fig. 25 shows a marine-type boiler of this nature as used for land purposes. It is arranged for firing with oil on the pressure system, or alternatively with coal in a mechanical stoker.

The path of the hot gases is indicated by arrows, and the superheater forms an integral part of the boiler. An economiser, made of similar steel tubes, is placed on the top of this boiler, where it saves space and at the same time receives the full benefit of the ascending flue gases.

Coal-fired boilers of this design have been installed in the Amsterdam Municipal Electric Generating Station and in the Walsall

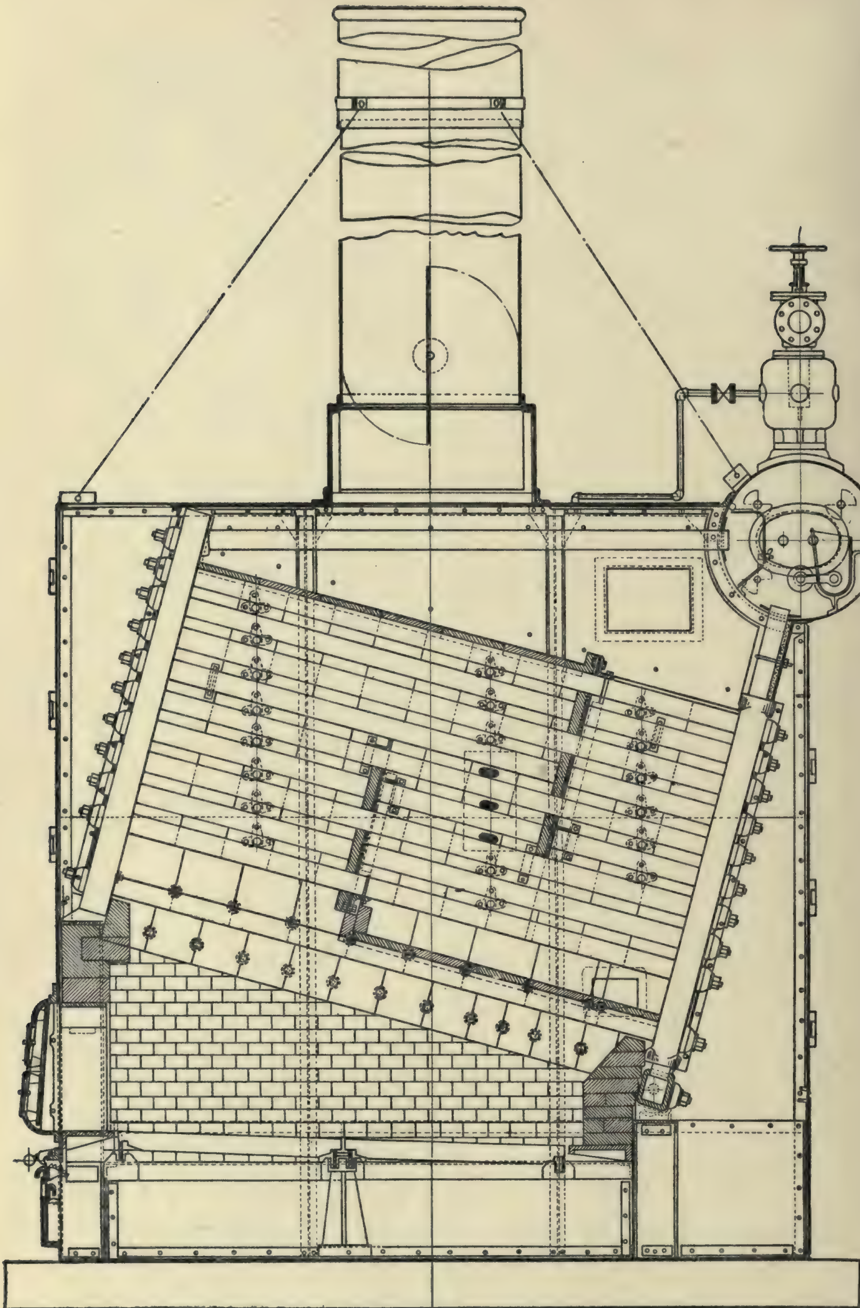


FIG. 24.—" Portable " Babcock boiler arranged for wood-firing.

Electric Power Station at Birchills in Worcestershire. The results are so good that detailed tests may be of interest.

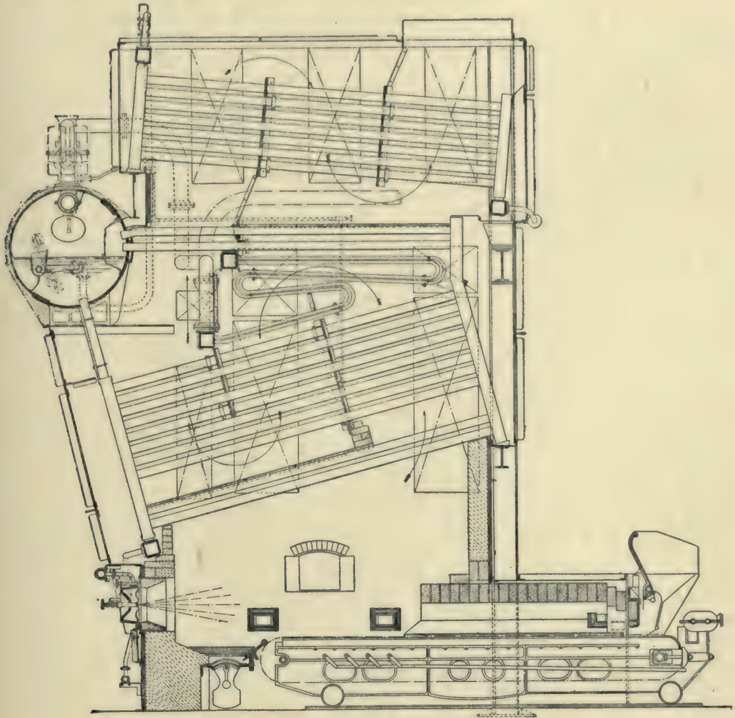


FIG. 25.—Babcock boiler for large power installations.

The general description of the boilers was as follows :—

Type and make (A. and B.) Water tube C.T.M., Babcock & Wilcox steel-cased boiler with integral superheater, superimposed economiser, and steel chimney. Mechanical chain grate stokers.

	Trial A. (Amsterdam.)	*Trial B.* (Birchills.)
Total grate area sq. ft. (excluding dead plate) .	210	140
Total effective heating surface, sq. ft. (boiler) . .	4963	3970
Ratio $\frac{\text{H.S.}}{\text{G.A.}}$	23·7	28·5
Heating surface, sq. ft. superheater	1813	1468
,, ,, ,, economiser	4086	2104
Chimney height above grate, ft.	94	80
Method of starting and stopping the test (A. and B.)	Flying start after boiler had been working several hours and conditions steady. Test finished with fire thickness, draught and water-level the same as at the start.	
Average thickness of fire, in..	4½	5½
Production of draught (A. and B.)	Cold air ejector.	

The main observations and deductions were as follows:—

	Trial A.	*Trial B.*
Date	March 24, 1915.	August 27, 1917.
Duration of trial	3 hrs. 46 mins.	8 hrs.
Coal burned.		
Description.	Washed nuts.	Walsall wood.
Net calorific value as fired, B.Th.U., per lb.	12,766	10,411
Fired, per hour, lb.	4489	3680
Per sq. ft. grate area, per hour, lb.	21·4	26·3
Per sq. ft. heating surface, per hour, lb.	0·9	0·925
Water evaporated.		
Total, per hour, lb.	40,359	25,448
Per lb. of coal as fired, per hour, lb.	8·99	6·92
Equivalent evaporation from and at 212° F., per lb. of coal fired, lb. (including superheater)	10·54	8·64
Ditto per sq. ft. heating surface, per hour	9·53	8·00
Steam pressure, lb. per square inch gauge	149	180
Degrees of superheat, ° F.	310	340
Rise of temperature in economiser, ° F.	120	92
Heat transferred to the water, per lb. of fuel as fired, B.Th.U. (boiler and superheater)	10,226	8,381
Economiser	1079	636
Thermal efficiency of boiler and superheater	80·10	80·50
Ditto of boiler, superheater, and economiser combined	88 56	86·6

The Niclausse Water-Tube Boiler.—This type of boiler differs from the Babcock and Wilcox in two main respects. The tubes are only connected to headers at one end instead of running between two sets of headers, and the water circulation is controlled so as to make it positive and in one direction only.

The boiler has a single steam and water drum running across the plane of the tubes and each vertical row of tubes has a header running up into the bottom of this drum. These headers are divided into two portions. The water flowing through the front portion is distributed to each tube by an inner tube which runs nearly to the closed end of the outer or evaporating tube. The outer tube connects by means of windows cut in its header end to the back portion of the header so that the water enters one way and the steam departs the other way in two distinct currents which ensures a positive circulation in only one direction.

A section of this header and the way it enters the drum is shown in Fig. 26, which, together with Figs. 27 to 29, are reproduced by kind permission of the Institution of Mechanical Engineers.

The circulation is further controlled by a number of baffles which can be seen in Fig. 27. The cold feed enters the trough C inside the drum and descends the headers as far as the baffle E. This baffle is omitted in one out of every four or five headers which are arranged to take their feed from the back part of the drum containing only purer and hotter water. All the headers are connected at the bottom to a horizontal box which is fed with this purified hot water. By this arrangement the upper tubes carry the relatively cold feed-water, whilst

the lower tubes which are most exposed to the heat of the furnace receive the feed at boiler steam temperature and free from any impurities which can cause deposit.

Fig. 28 shows a Niclausse land-type boiler erected at the Corporation Electricity Works of Southend-on-Sea, England. It is designed for a

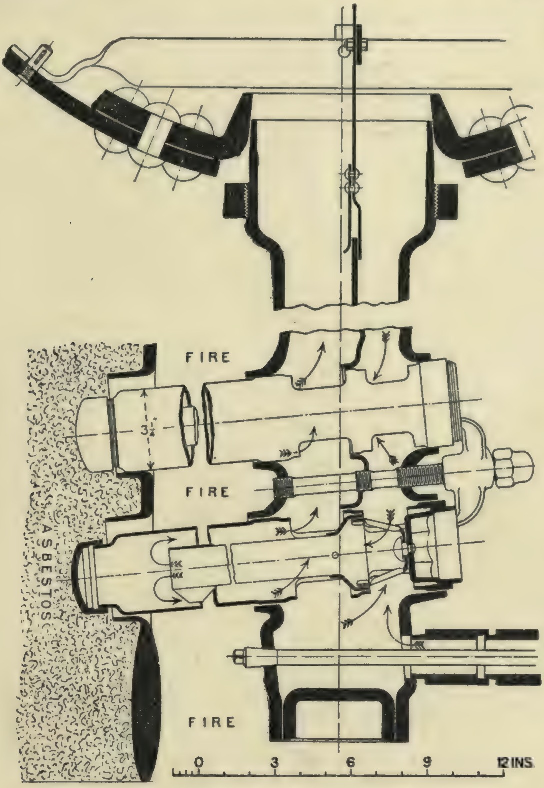


FIG. 26.—Niclausse boiler. Detail of header.

duty of between 25,000 and 35,000 lb. of steam per hour and is fitted with an economiser and mechanical stoker, and also a superheater made of serpentine sections interpolated and coupled to two headers, one at the entrance and the other at the steam exit. A break is made in the nests of water tubes to leave room for the superheater. The air supply is furnished by a forced-draught fan working at low pressure, and placed underneath the stokehole. The outside diameter of the evaporating tubes is about $3\frac{5}{16}$ in. and their slope is 1 in 10.

A modern marine-type of Niclausse boiler is shown in Fig. 29. Here it is arranged for oil firing, but it is equally suitable for solid fuels. It will be noticed that two downtake pipes are fitted for the purified hot feed water, one on each side of the boiler, instead of using a number of headers for this purpose as in the most recent land type.

There is no superheater, and the economiser is placed in the flue uptake above the boiler. The dotted lines shown across the tubes in the front elevation represent a number of plain open-ended tubes, laid between the evaporating tubes, which act as baffles and increase the path of the hot gases. This boiler has 13 headers, and 456 tubes, $2\frac{3}{8}$ in. outside diameter (66 mm.). Of these 300 are upper tubes and 156 are lower tubes fed by the hotter water only. The headers are 7 ft. 6 in.

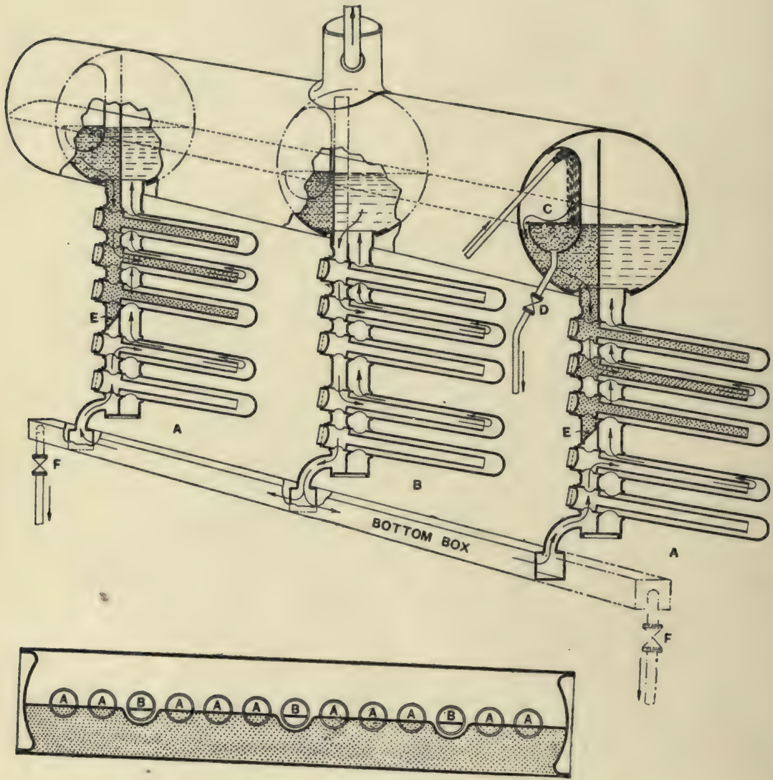


FIG. 27.—Diagram of water circulation in Niclausse boiler.

high and the tubes project 6 ft. $7\frac{1}{4}$ in. outside them. The economiser has 780 tubes 4 ft. long.

Some interesting trials were taken on this boiler, of which the following is a summary. These show the very high efficiency obtainable with controlled circulation. Trial A was taken by representatives of the French Admiralty and Trial B by Captain H. Riall Sankey and a committee of British engineers. Both trials took place in France on the same boiler.[1]

[1] See *Proc. Inst. Mech. Eng.* 1914, pp. 507 *et seq.*, and *The Engineer*, 1914, vol. cxvii, pp. 280 and 542.

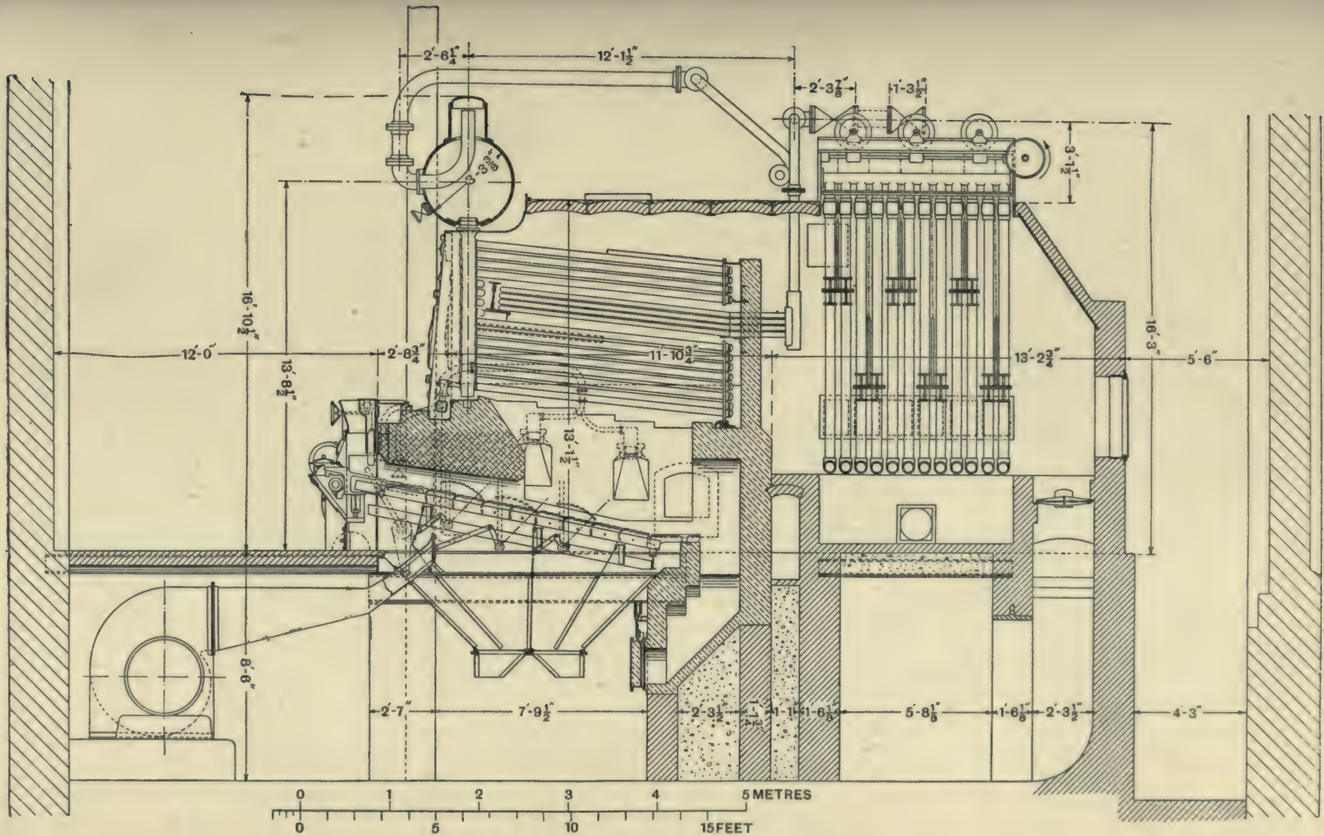


FIG. 28.—Niclausse boiler, land type.

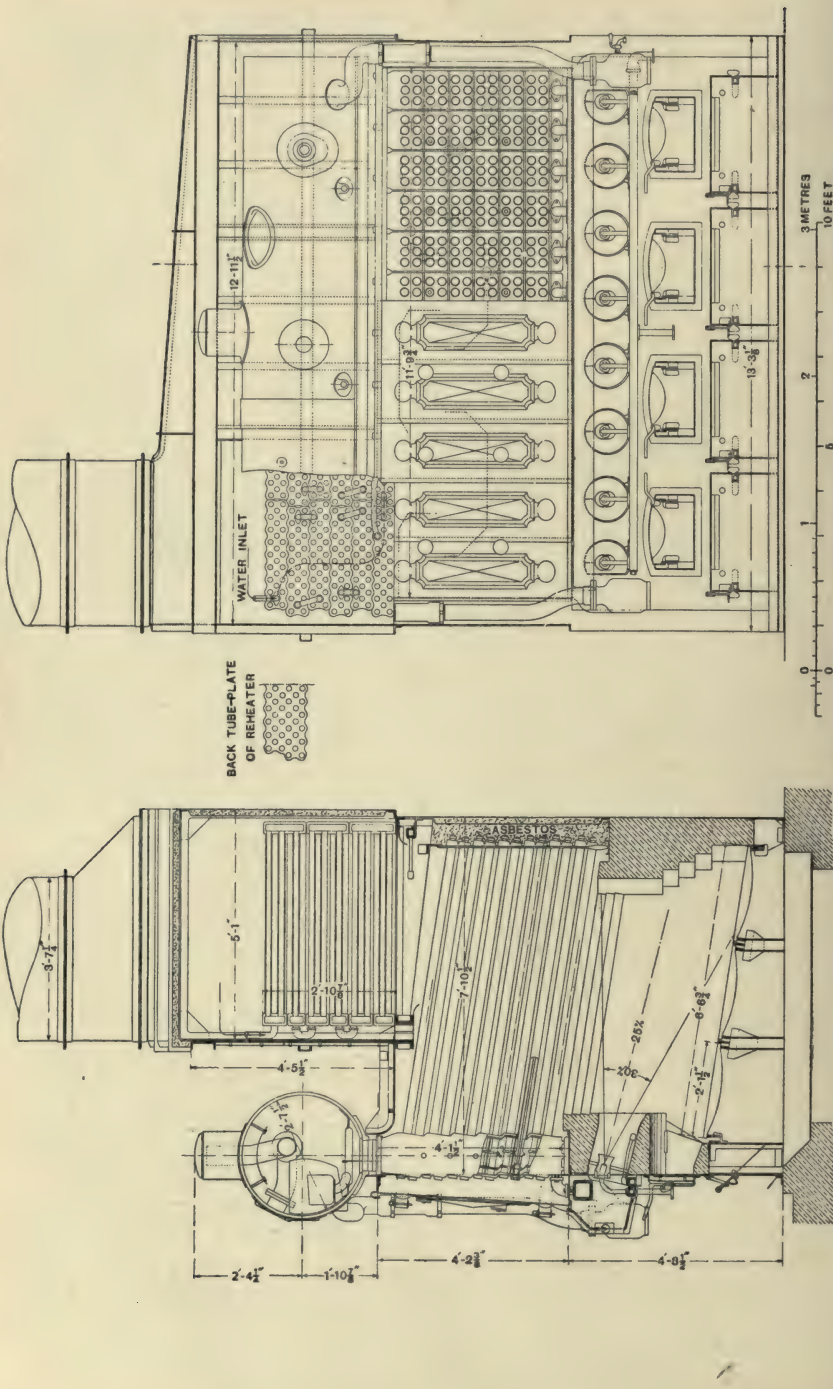


FIG. 29.—Niclausse boiler, marine type.

In addition to the above dimensions, the following were the particulars of the boiler when under test—

Total grate area	77½ sq. ft.
Heating surface of outer tubes	1870 ,,
,, ,, economiser	1285 ,,
Total heating surface	3155 ,,
Ratio of outer heating surface of the tubes (neglecting the header surface) to the grate area	40·67 to 1
Production of draught	Natural.

The main observations and deductions were as follows :—

	Trial A.	*Trial B.*
Authority	French Admiralty.	Captain Sankey.
Date	September, 1913.	February, 1914.
Duration, hours	6	2
Fuel.		
Description.	Briquettes d'Anzin.	Briquettes d'Anzin.
Net calorific value as fired B.Th.U., per lb.	14,000	14,000
Fired, per hour, lb.	1510	1425
Per sq. ft. grate area, per hour, lb.	19·5	18·4
Per sq. ft. total heating surface, per hour, lb. (boiler and economiser)	0·48	0·453
Per sq. ft. heating surface, per hour, lb. (boiler only)	0·807	0 762
Water evaporated.		
Total, per hour, lb.	16,620	15,140
Equivalent evaporation from and at 212° F., lb.	19,930	18,696
Ditto, per lb. of coal fired	13·19	13·12
Ditto, per sq. ft. heating surface (boiler and economiser)	6·3	5·9
Ditto, per sq. ft. heating surface (boiler only)	10·65	10·0
Steam pressure, lb. per sq. in. (gauge)	256	213
Temperature of feed to economiser, ° F.	77	41
Ditto, to boiler, ° F.	215	192
Thermal efficiency of boiler and economiser	$\frac{13 \cdot 19 \times 966}{14{,}000} = 91 \cdot 0$	$\frac{13 \cdot 12 \times 966}{14{,}000} = 90 \cdot 5$

The Stirling Boiler.—This boiler, much used in America as well as in Britain, differs from the other three types of vertical tube boilers in two main respects.

The tubes are curved instead of being straight and the mud drums are less in number than the steam and water drums above. The smallest sizes have two steam drums and one mud drum. The intermediate sizes, three steam drums and one mud drum. The larger sizes, which are made up to 10,000 sq. ft. of heating surface, three steam drums and two mud or water drums.

A modern example of the three-drum type with superimposed economiser is shown in Fig. 30.

Fig. 31 shows a drawing of the standard five-drum type equipped with a chain-grate stoker and superheater. The same boiler with a suitable furnace is equally adaptable for solid, pulverised, or liquid fuels and waste gases. The hot gases are guided by means of the baffles, which can be clearly seen, along and through each nest or bank of tubes, and their exit is controlled by a wing damper.

The circulation of the water, though free, is particularly good. The feed enters the steam drum furthest away from the fire and is evenly distributed along the whole length of the drum by means of the U-shaped baffle shown. There is no water connection between this and

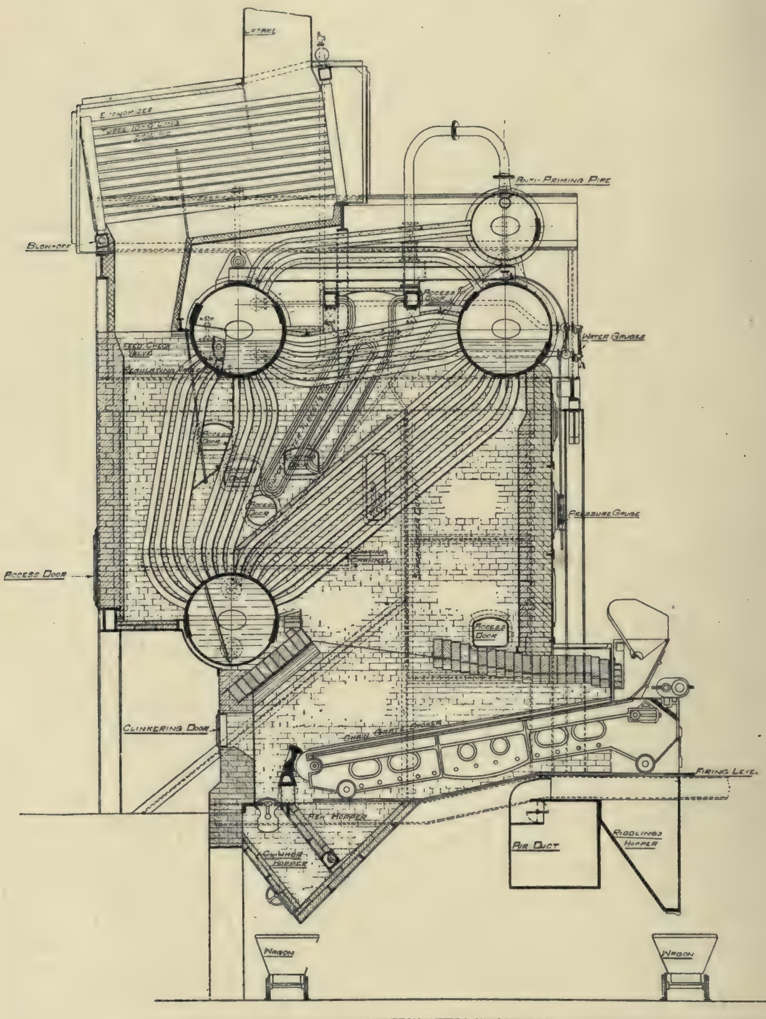


FIG. 30.—Stirling three-drum boiler.

the next steam drum, a feature which is peculiar to this boiler, and as a consequence the feed passes down the rear bank of tubes and deposits any sediment in the rear mud drum at the bottom. The bulk of the water rises up the next set of tubes and then travels in a circle round the

remaining pair of banks and the short tubes connecting the two front steam drums.

In order to obtain a somewhat similar effect in the three-drum marine type, the rear steam drum and the mud drum are divided into two separate portions. The feed enters a closed box which covers only half the rear bank of tubes, the remainder of which are free to act as risers.

In this way half the rear bank in the top drum and the whole of

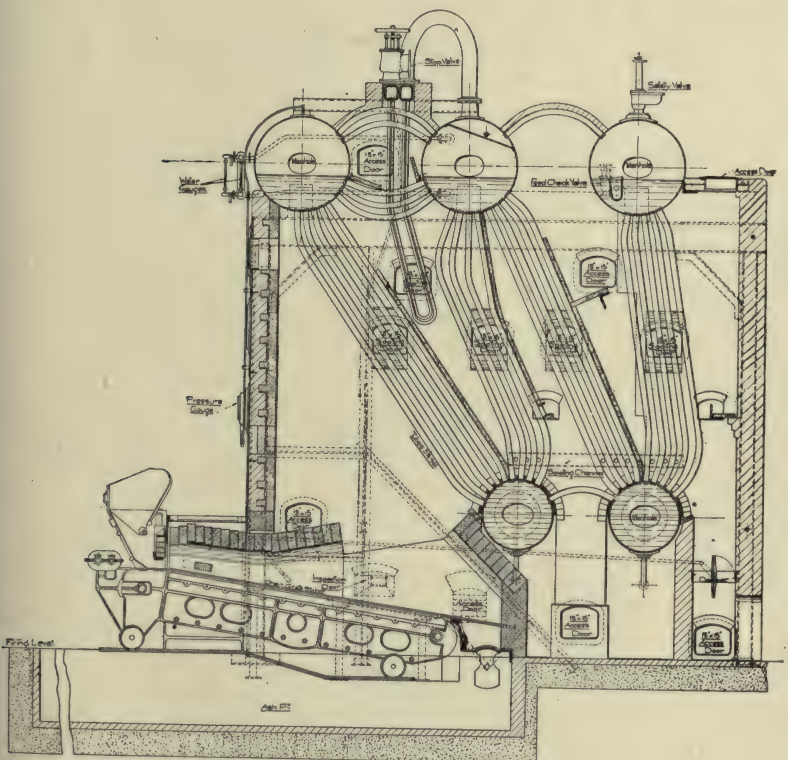


FIG. 31.—Stirling five-drum boiler.

it in the rear drum is isolated from the remaining tubes of the boiler. All the water has to pass down and upwards again before circulating round the hotter parts of the boiler.

The Woodeson Boiler.—In this boiler there are a number of elements, usually three, each consisting of a steam drum and a water drum connected by a nest of straight tubes. These tubes are arranged in groups, and are expanded into flat discs pressed out of the sides of the drums. On the top of the steam drums a manhole is placed over each group of tubes, so that any one tube can be conveniently withdrawn and replaced.

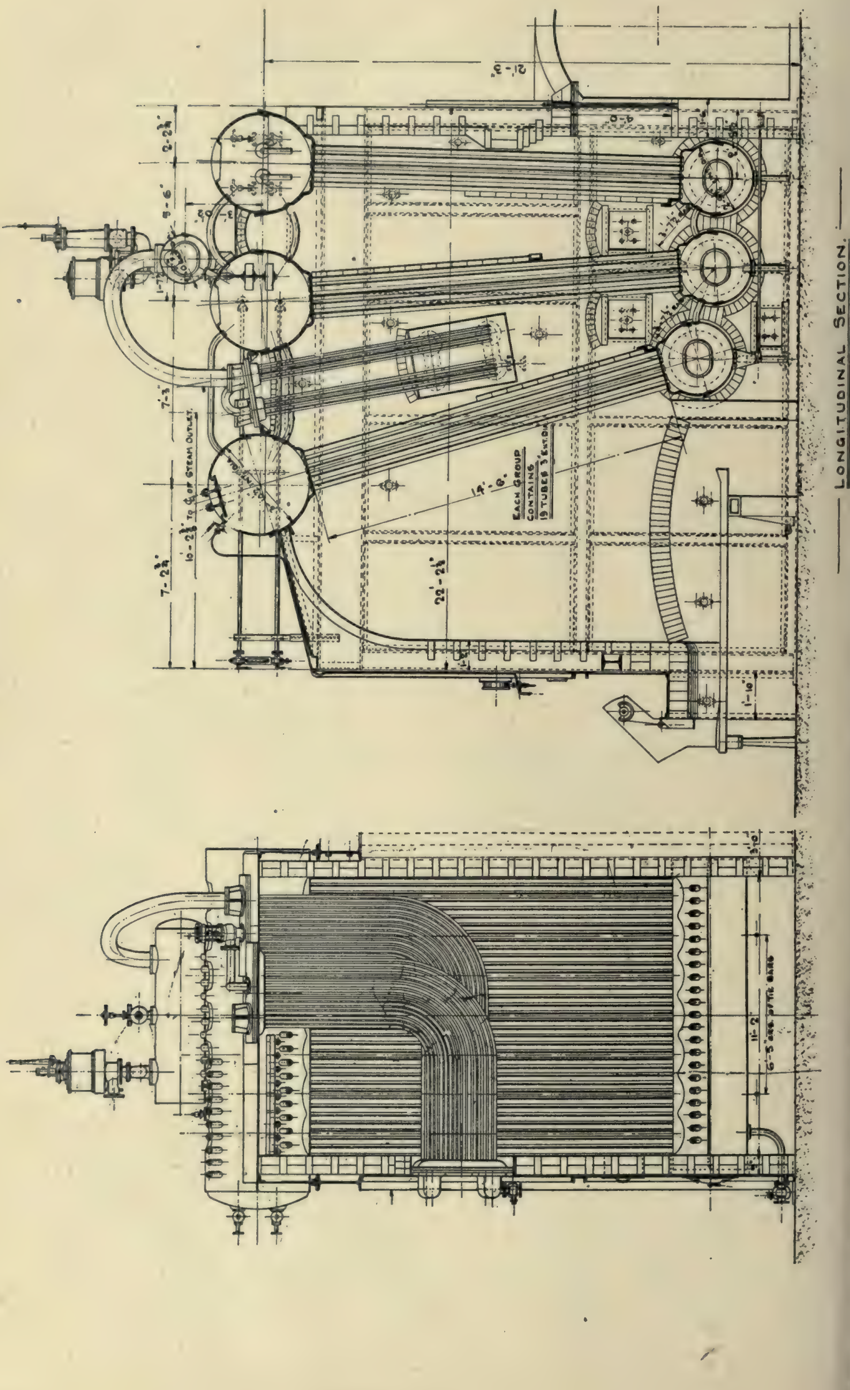


FIG. 32.—Woodeson vertical water-tube boiler.

The elements are inter-connected with short lengths of pipes, both above and below, and the whole unit is suspended on girders and left free at the bottom for expansion.

A steam dome is arranged over the steam drums to ensure dry steam, and the superheater, consisting of a number of L-tubes joined at their lower end, is placed between the first and second element.

The hot gases are guided by means of firebrick baffles up and through the first nest of tubes, then across and down the superheater and towards the two front water drums. The third baffle in front of the rear nest of tubes deflects the gases once more before they pass out of the flue exit at the back of the boiler.

The feed enters the rear steam drum and passes downwards to deposit any solid matter as mud in the bottom drum, before it is distributed between the front two elements. The boiler shown in Fig. 32 has a heating surface of 4750 sq. ft. and a grate area of 79 sq. ft.; the ratio being 60 to 1. The thermal efficiency of these boilers often exceeds 80 per cent. without economisers, and the largest size, with 6100 sq. ft. of heating surface, has a normal evaporation of 30,000 lbs. of water per hour from and at 212° F.

The *Thompson* boiler, made under the "Sinclair" patents, is similar to the Woodeson in design and principle and differs mainly in constructional details.

The Nesdrum Boiler.—A drawing of this type of boiler is given in Fig. 33, and it will be seen that it differs from the Thompson and Woodeson in having a series of nests of tubes ending in small cylindrical drums, each element being inter-connected with short pipes, and several nests being arranged side by side in rows.

The water circulation is similar, but all the steam is collected in one large steam and water drum, shown in cross section between the superheater and the top of the rear nest, which runs the whole width of the boiler. The boiler as shown is fitted with apparatus for gas-firing.

EXTERNALLY-FIRED WATER-TUBE BOILERS (SMALL TUBES)

The small tube, or "Express" type of boiler as it is often called, is applied almost exclusively to marine work. It is particularly compact for its duty and lends itself to the rapid evaporation of steam when required.

Two boilers of this class which have now much in common are the *Yarrow*, made by Messrs. Yarrow & Co., Ltd., Scotstoun, Glasgow, the *Thornycroft*, by Messrs John I. Thornycroft & Co., Ltd., Woolston, Southampton. The *White-Foster*, made by the Babcock & Wilcox Co., is also similar in its main design to these two makes.

The Yarrow Water-Tube Boiler.—This type has always consisted of a central steam and water drum from the under or water side of which two banks of tubes, inclined at about a right angle to one another, have terminated in two smaller water drums placed one each side of the fire grate. These water drums were more or less "D" shaped in section as can be seen in the photograph on p. 92, but the latest types as shown in Fig. 34 have cylindrical drums throughout.

When a superheater is included the bank of tubes on that side is diminished to give approximately the same total heating surface on each side of the boiler. The superheater has only one drum, which is accessible and easily constructed and cleaned. The uptake is divided into two portions and the superheater side is fitted with a damper.

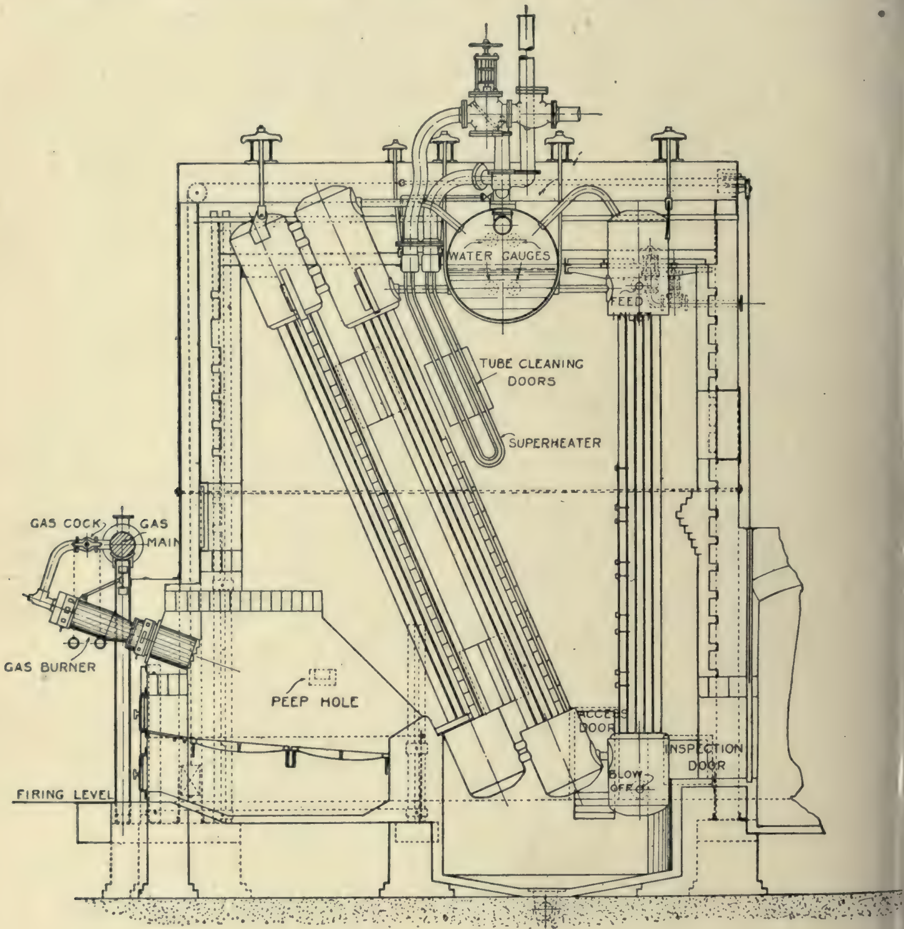


FIG. 33.—Nesdrum vertical water-tube boiler.

This enables the superheater to be shut off when raising steam or when the main engines are suddenly eased or stopped. At the same time the output of the boiler is diminished just when a reduced supply is needed.

The feed, which formerly entered the water drum, is now brought in

along the bottom of the main steam and water drum. All the water tubes are straight except those nearest the fire. The slight curvature here enables these tubes, which have thicker walls than the others, to be fitted in without increasing the size of the drums. The riveted joints of the steam drum are protected by baffles from direct contact with the hot gases. The steam drum is often fitted with steel trays (not shown), to carry zinc slabs; when these zinc slabs become oxydised and brittle, the loose pieces remain in the trays and do not get into the tubes. The zinc prevents corrosion and a pair are sometimes fitted high up in

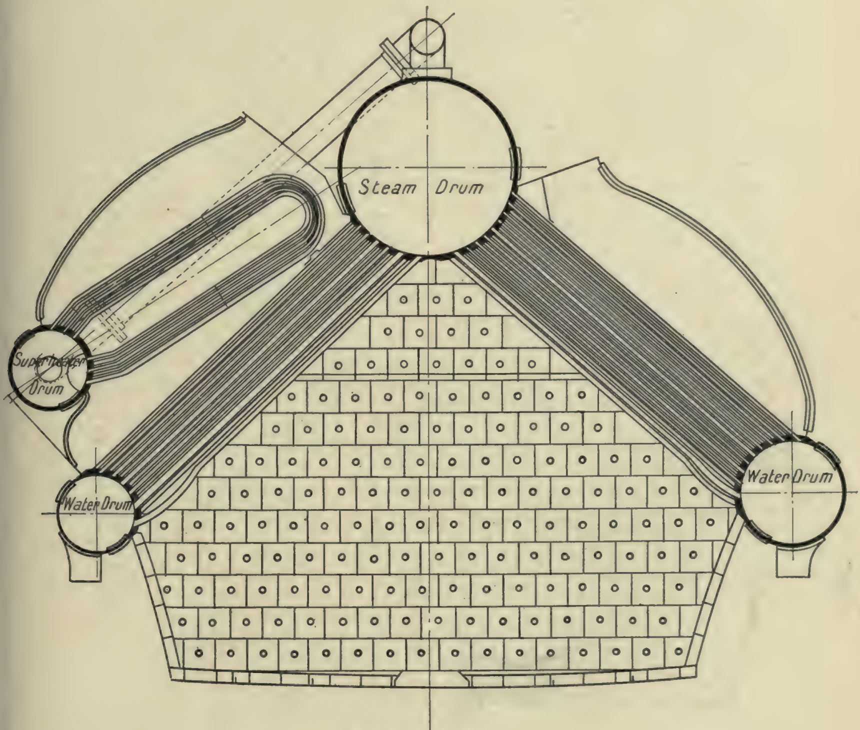


FIG. 34.—Modern Yarrow water-tube boiler.

the main drum to prevent corrosion when the boiler is filled to the top, a usual practice in the British navy when boilers are laid up.

As the hot gases pass through the bank of tubes their volume is reduced due to their lower temperature. The area between the outside tubes of the bank is therefore reduced by angle baffles so that the distribution of the furnace gases over the whole of the bank is more uniform.

The heating surface of the boiler, shown in Fig. 34, is 5900 sq. ft. in the generating tubes and 1440 sq. ft. in the superheater, making a total of 7340 sq. ft. for the whole boiler.

The Thornycroft Water-Tube Boiler.—The present-day Thornycroft boiler, shown in Fig. 35, is the outcome of long practical experience of the firm of John I. Thornycroft & Co., Ltd., dating back to 1885. The "Speedy" type and its successor the "Daring" type had curved water tubes, which discharged into the steam space of the steam and water drum. These were followed by the "Thornycroft-Schulz" and the modified "Speedy" types, in which most of the water tubes were drowned. These four types can be seen in diagram form in Fig. 36.

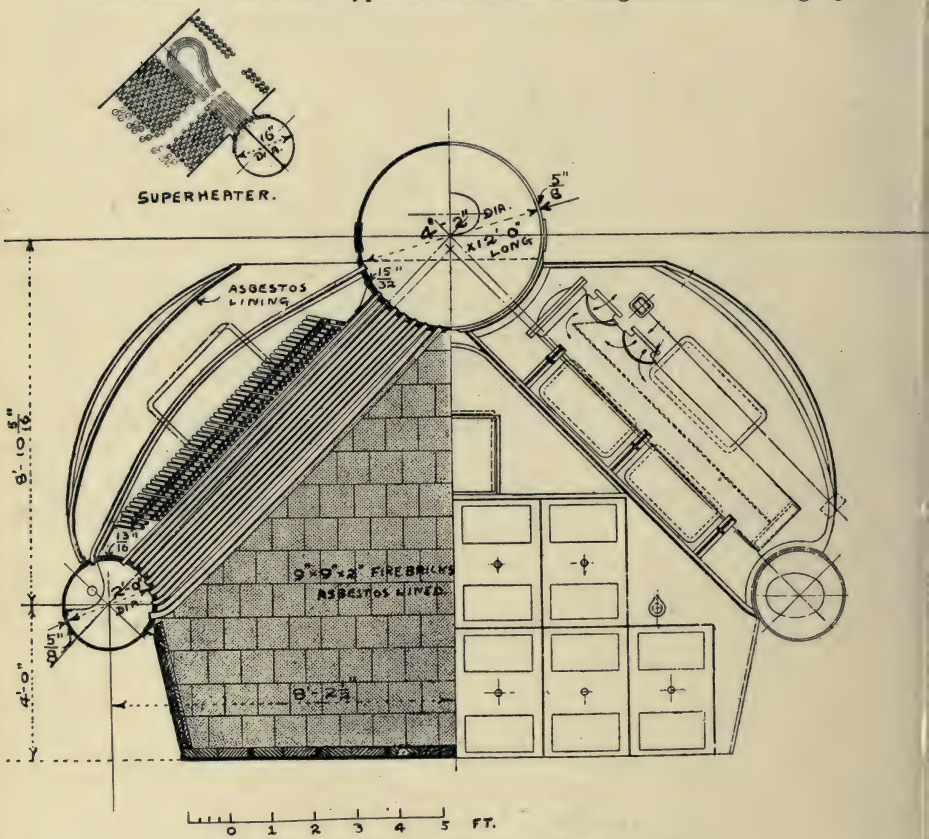


FIG. 35.—Modern Thornycroft water-tube boiler.

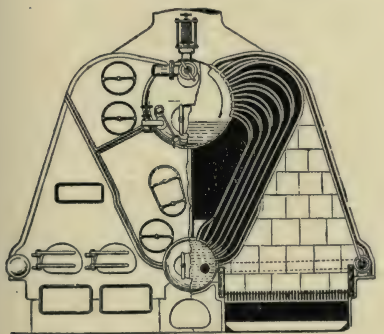
The modern type has practically straight tubes which can be sighted throughout their length; it can be designed for coal, oil, or wood fuel.

The feed comes into a partitioned space in the water drums and passes to the central drum through two rows of tubes, outside the superheater, which practically amount to feed-water heaters. All the tubes are drowned as in the Yarrow boiler, and have an external diameter of $1\frac{1}{8}$ in., except those nearest the fire which are $1\frac{3}{8}$ in. diameter. Both sides are balanced and equal, and both contain superheater tubes of the special shape shown in the detail.

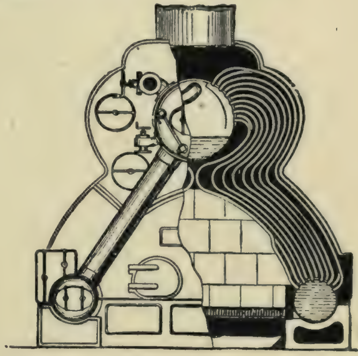
Tests on such a boiler when fired with oil are given in the following table :—

TESTS ON THORNYCROFT BOILER (FIG. 35).

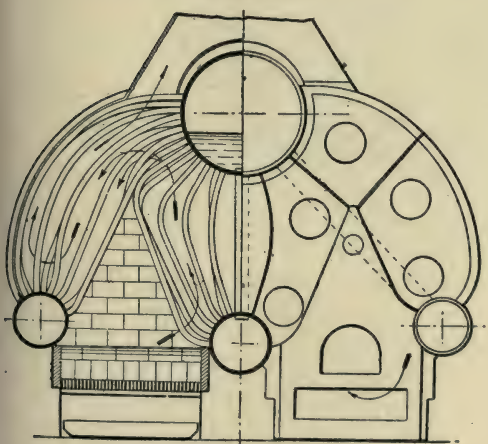
Date		November, 1914.	
Duration of trial	2 hours.	2 hours.	2 hours.
Heating surface in sq. ft.—			
Generating tubes		5620	
Superheater tubes		1370	
Feed tubes		940	
Total		7930	
Steam pressure (gauge) in lb. per sq. in.	225	226	250
Oil burners in use	2	6	9
Lb. of oil burnt per hour	2180	6510	10,255
Ditto, per sq. ft. heating surface, per hour	0·275	0·82	1·292
Temperature of feed, ° F.	46	44	45
Lb. of water evaporated, per hour	28,425	79,200	118,150
Ditto, from and at 212° F.	35,000	97,700	145,900
Ditto, per lb. of oil fuel	16·03	15·00	14·23
Ditto, per sq. ft. heating surface	4·42	12·30	18·39
Thermal efficiency (taking cal. value of the oil at 19,000 B.Th.U. per lb.)	$\frac{16·03 \times 966}{19,000} = 81·5$	79·0	75·0



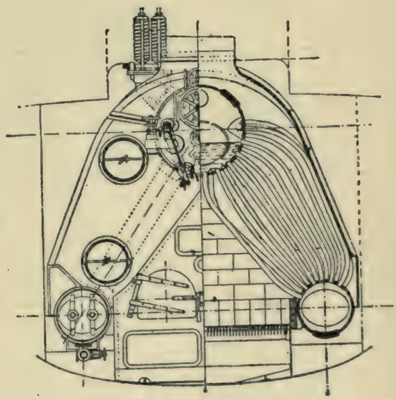
"Daring" type.



"Speedy" type.



"Thornycroft-Shulz" type.



Modified "Speedy" type.

FIG. 36.—Former designs of Thornycroft boilers.

The enormous evaporative power of the Express water-tube boiler compared to the cylindrical type should be noted. The combined space of this boiler was 1010 cub. ft., and a Scotch marine boiler which occupied $10\frac{1}{2} \times 10\frac{1}{2} \times 10\frac{1}{2} = 1158$ cub. ft., is shown in tests recorded on p. 74 to have reached an equivalent evaporation of only $989 \cdot 3 \times 11 \cdot 19 = 11{,}300$ lb. of water.

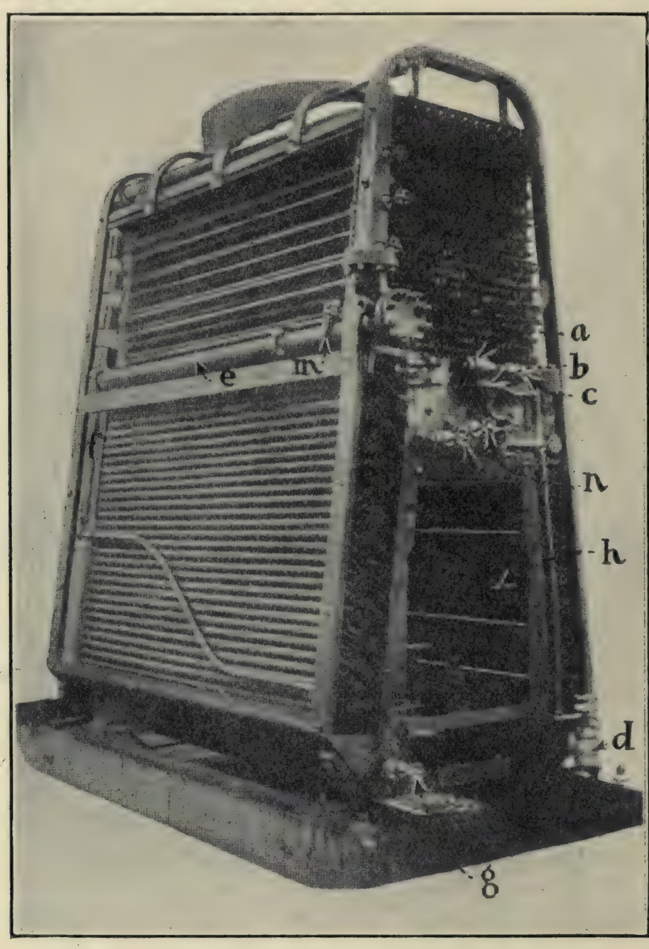


FIG. 37.—Talbot boiler. Front of boiler, casing removed.

a, Automatic regulator value; *b*, Oil valve; *c*, Hand valve; *d*, Main check valve; *e*, Main steam pipe; *f*, Separator; *g*, Separator drain; *h*, Combustion space; *m*, Atomiser pipe; *n*, Burner valves.

The American Talbot Boiler.—The Talbot Boiler Co. of New York now make an Express internally-fired water-tube boiler, which is essentially of the counter-flow type. Artificial circulation is used, the water being maintained at a high velocity by means of a pump, and there is no steam and water drum. Figs. 37 and 38 show

the appearance of the front and rear ends of a Talbot boiler with the outer casing removed. The capacity is 15,000 lb. of steam per hour with $\frac{1}{4}$ in. of water draught or double that amount with slightly over 2-in. water draught.

The tubes throughout the boiler are all alike, as can be seen from Fig. 39. They are built up into units which consist of steel headers at one side only. The header is divided into two compartments, into

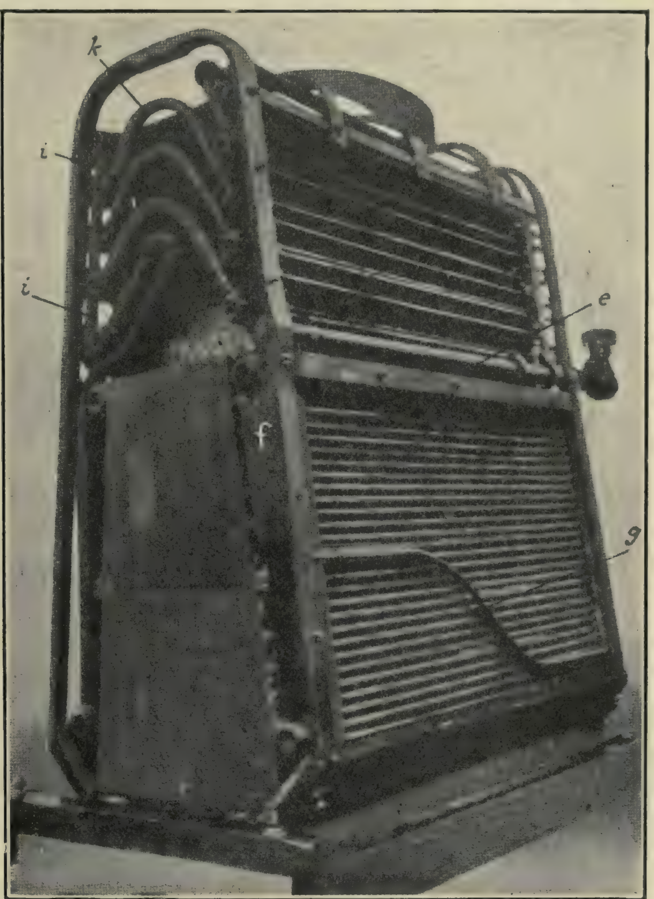


FIG. 38.—Talbot boiler. Rear of boiler, casing removed.
f, Separator; *e*, Main steam pipe; *g*, Separator drain; *i*, Feed pipe; *k*, Trap.

one of which is screwed open-end "Field" tubes and into the other closed evaporating tubes which surround the inner tubes. Five of these units with horizontal headers are placed above the combustion chambers and act as a feed-water heater, whilst on each side is arranged another set of tubes with a vertical header. All the headers are joined up in series with trap connections, as can be seen in Fig. 38.

The feed is pumped in through a check valve at the side of the boiler and passes through a pipe with a regulator valve straight to the top nest of tubes. The circulation through this nest may be seen marked by arrows in Fig. 39. It will be noticed that the tubes are arranged in pairs. In the first pair the water flows through the inner tubes and back through the outer ones, whilst in the next pair it flows through the outer tubes first and back through the inner ones to the header.

The feed passes from header to header downwards until it reaches the left-hand vertical nest, through which it flows in the same way to the bottom of the boiler. At this point it crosses the back of the boiler through a wall which is partitioned to maintain a high velocity,

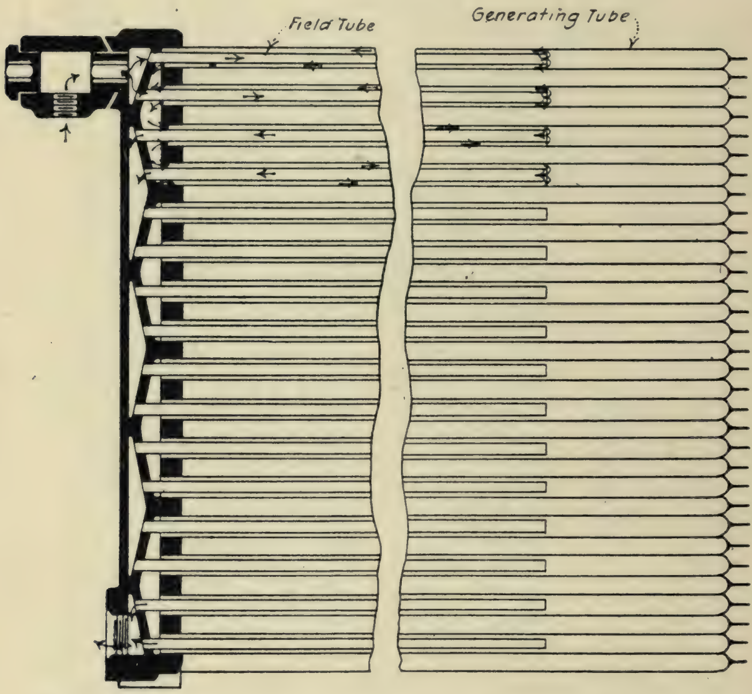


FIG. 39.—Diagrammatic section through header and tubes of Talbot boiler. (Arrows show course of water circulation.)

and flows upwards in the form of steam through the set of tubes on the right-hand side of the furnace, which act as a superheater. The total length of the water passage is nearly 800 ft., and the velocity of flow is about 500 ft. per minute in the first set of tubes to 12,000 ft. per minute in the last passages, which contain only steam. The friction of the circulation is considerable, and the average evaporation over all the tubes is 8 lb. per sq. ft. at normal load.

The boiler requires thermostatic control of both fuel and water supply, so that it is practically confined to liquid fuels. This control

is somewhat ingeniously worked by the expansion and contraction of one of the sets of tubes which is magnified by a system of levers and operates the regulating valves.

The artificial circulation on the counterflow principle makes for rapid and speedy evaporation. In one test the equivalent evaporation

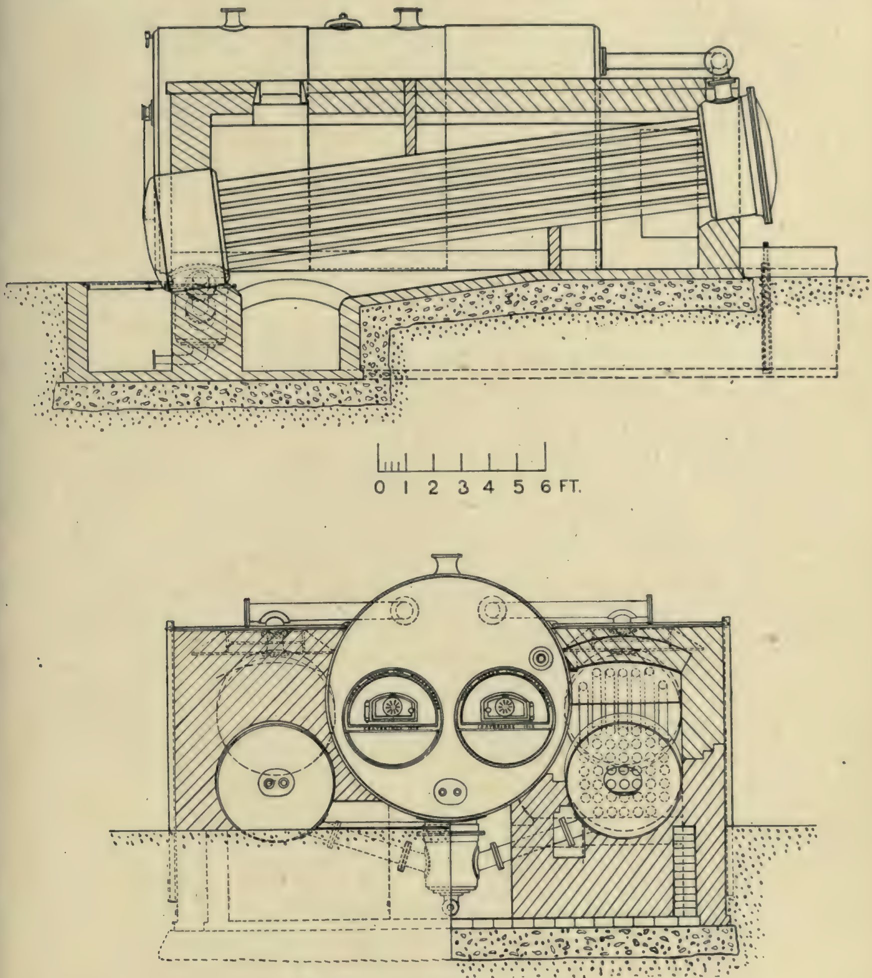


FIG. 40.—Hudson boiler.

per lb. of oil was 16 lb. of steam per hour, with 200° F. of superheat. With oil of 18,260 B.Th.U. per lb. calorific value this represents a thermal efficiency of $\frac{16 \times 970}{18,260} = 85$ per cent.

Further details and criticisms of this boiler are given in a paper by

Paul A. Talbot, presented to the American Society of Mechanical Engineers,[1] and, by the courtesy of their secretary, Figs. 37–39 have been reproduced from their Transactions.

INTERNALLY-FIRED WATER-TUBE BOILERS

The Hudson Boiler.—A design which combines desirable features of the cylindrical and water-tube types is patented by Messrs. Thomas Hudson & Co., of Coatbridge. The boiler, as can be seen from Fig. 40, consists of a Lancashire type two-flue cylindrical drum, flanked on each side by a nest of large-bore water tubes. These water tubes are set at a slope of about 1 in 8, and are connected through the

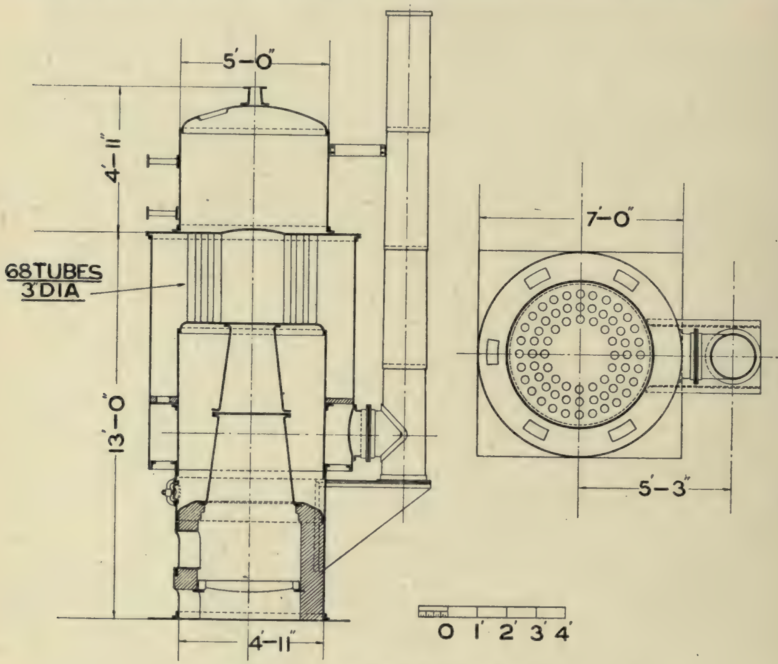


FIG. 41.—Hudson vertical boiler.

lower headers to a mud drum underneath the front end of the cylinder. The rear headers each discharge the steam generated in the water tubes into the steam space at the back of the boiler drum. The hot gases pass from the fires in the internal flues to the back of the cylindrical element, and are then divided to right and left by the brickwork setting into the side flues, down which they flow along and over the water tubes as well as past the sides of the drum. From this point they pass along the main flue to the economiser or the chimney.

The feed water enters the front of the drum, to the bottom of which it gravitates. Part of it remains in the cylindrical portion and the

[1] *Trans. Am. Soc. Mech. Eng.* 1916, vol. xxxviii. pp. 1117–1140.

remainder flows through the mud drum to the lower header of the water tubes. The water is rapidly heated as it passes up the tubes in counterflow to the hot gases and discharges as steam into the back end of the boiler proper.

The design is made up to 18 ft. × 9 ft. diameter in the cylindrical

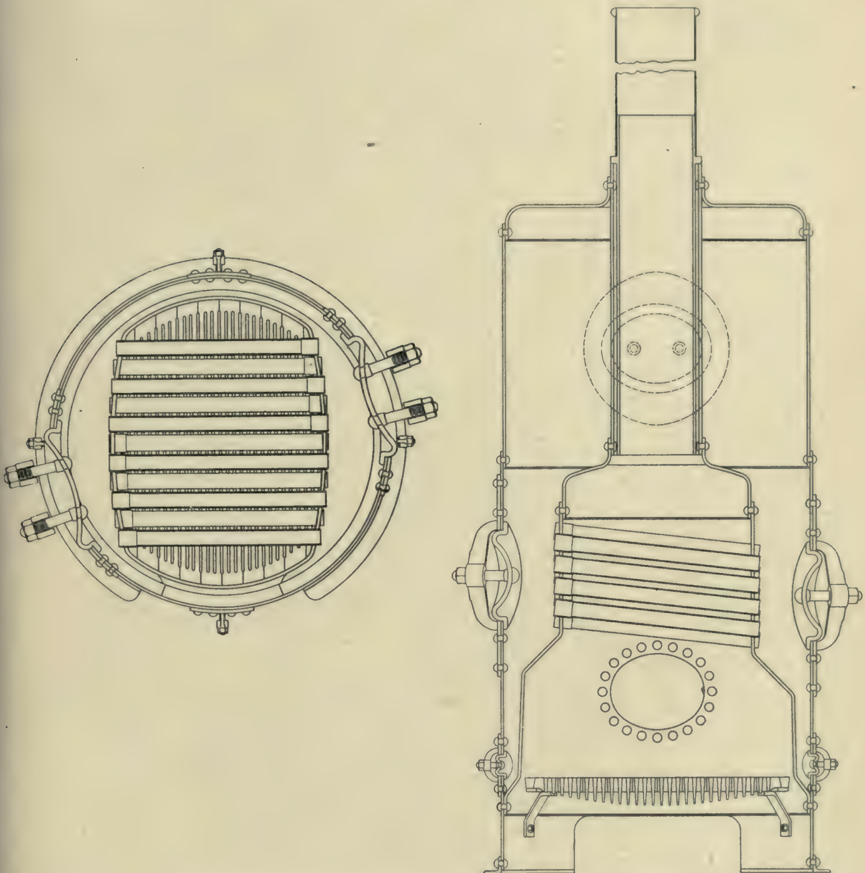


FIG. 42.—Spencer-Hopwood vertical water-tube boiler.

element and 57 4-in. tubes in each nest, with an equivalent evaporation of 17,000 lb. of water per hour. The smaller sizes are made single-flued, like the Cornish type. Fig. 40 shows a boiler 16 ft. × 8 ft. diameter, with 45 4-in. tubes in each side, which has an equivalent evaporation of about 12,500 lb. of water per hour. The heating surface of this boiler is 2050 sq. ft., and the grate area 44 sq. ft., the ratio being 46·6 to 1.

The same firm also make a **vertical water-tube boiler with vertical tubes**, which is shown in Fig. 41. It consists of two

cylindrical elements joined by straight vertical water tubes. The firebox is brick lined, instead of being surrounded by water, and the gases pass through the tapered uptake and through the water tubes. They then travel downwards between the shell and the outer casing, which is lined with asbestos, and through suitable portholes to the chimney. The two elements are connected by an external circulating pipe on which the feed valve is placed, and the hot water from the top cylinder mixes with this incoming feed, and raises its temperature.

An example **of a vertical water-tube boiler with horizontal tubes**, as made by Messrs. W. H. Spencer & Co., of Hitchin, is shown in diagrammatic form in Fig. 42. It is known as the "Spencer-Hopwood" boiler. In this case the space above the fire is filled with a nest of horizontally inclined water tubes, which add to the heating surface of the common vertical boiler. By arranging two manholes, as seen in the plan, any tube is easily accessible for cleaning or for removal. The tube plates, which are slightly concave, are made in one with the firebox shell so that no riveted joint is exposed to the fire, except at the crown plate. The largest standard size in which this boiler is made is 12 ft. high by 5 ft. 6 in. diameter, with an equivalent evaporation of about 2500 lb. of water when working at 100 lb. per sq. in. gauge-pressure. The ratio of heating surface to grate area varies from 10 to 1 in the smallest sizes up to 20 to 1 in the largest.

CHAPTER III

THE PRINCIPLES OF STEAM GENERATION

THE construction of apparatus to achieve any particular purpose falls very naturally into three stages. The *principle* upon which the apparatus is going to act must first be decided upon. Its *design* must then be worked out, and thirdly its *construction* must be carried out in accordance with that design. The three stages are interdependent and control one another. A knowledge of all three is essential to become a good designer.

The principles, based partly on theory and experiment, but in the case of boilers mainly on practice, decide the general arrangement or "lay-out" of the apparatus. As any definite arrangement becomes common it is referred to as a type. The design is controlled by constructional considerations, by the properties of the materials used and by such practical requirements as reliability, accessibility both for cleaning and for repairs, convenience in working, and initial cost.

The actual construction requires that knowledge of handling and shaping of material which can only be acquired in a practical training.

It has already been pointed out that the modern boiler, as typified by most of the examples which have just been considered, is, to a considerable extent, the outcome of natural evolution, the survival of the fittest, as Professor Dalby has aptly put it. The process has been costly, but the results are good.

It should not be inferred from this that a large amount of research work has not been done, both by firms and by individuals, on problems directly connected with boiler design. On the contrary, more than seventy years ago experiments were made on a locomotive boiler to find how the evaporation per sq. ft. of heating surface declined the further the hot gases travelled from the firebox.[1]

Since that time the fundamental question of Heat Transmission across the heating surface of a boiler has occupied the attention of many scientists and engineers. Professor Dalby, in his important paper on "Heat Transmission," gives a list of over 400 references up to the year 1909.[2] It is only possible here to consider a few experiments which emphasise valuable points in the principles underlying the design of actual boilers.

[1] Dewrance and Woods in 1842, *Trans. Inst. Nav. Arch.* vol. iii. p. 122.

[2] *Proc. Inst. Mech. Eng.* 1909, p. 921.

A series of three articles by J. G. Hudson on "The Heat Transmission in Boilers,"[1] gives, with one important exception, the information available in 1890. The general conclusions he arrived at are borne out on the whole by our present knowledge. They are reprinted here by kind permission of the editor of *The Engineer.*

"A comparison of a number of the most trustworthy recorded data of boiler tests, with formulæ based on the principles it suggested, resulted in the following propositions as to the laws governing the transmission :—

"*As regards surface exposed to the fire.*

"1. That the transfer of heat takes place principally by radiation from the fuel and flame, that by convection being in most cases comparatively unimportant.

"2. That a large extent of firebox surface in proportion to fuel burnt results in the absorption of a correspondingly large proportion of the heat developed, but in a small absorption per unit of surface, and *vice versâ.*

"3. That an excessive air supply, by reducing the temperature of combustion, reduces the absorption.

"*As regards surface exposed only to gases.*

"4. That the gases transfer their heat almost wholly by convection, the activity of which bears some proportion to the speed with which they traverse the absorbing surface; a transmission per degree directly proportional to the square root of the speed being found to agree closely with actual results.

"5. That the transmission per degree difference is greater at high than at low temperatures, a rate proportional to the mean between the absolute temperatures of the gases and the water being found to agree closely with the actual results.

"*As regards all boiler heating surface.*

"6. That as long as the water, particularly if in a state of ebullition, is in contact with one side of the surface, and the metal composing the latter is solid—*i.e.* not laminated—the temperature of even the hotter side of the surface—viz. that next the fire or gases—can only exceed that of the water by an amount so trifling in relation to the whole difference that for most practical purposes it may be taken as identical with that of the water."

Hudson was one of the first to point out the effect of the speed of the hot gases in boilers on the transmission of their heat, though his statement that it varies as the square root of the speed is not in accordance with Professor Osborne Reynolds's deduction from theoretical considerations that the amount of heat transmitted is a linear function of the speed of the fluids.[2] Hudson, at the time he wrote, was probably unaware of the existence of this paper, for he is careful to explain that the facts and data at his disposal were not sufficiently

[1] *The Engineer*, 1890, vol. lxx. pp. 449, 483, and 523.

[2] "On the Extent and Action of the Heating Surface for Steam Boilers," *Manchester Lit. and Phil. Soc.*, 1874.

complete to justify the conclusions drawn from them being put forward as sufficiently proved, but rather as highly probable approximations.

Professor John Perry, in his book on the steam engine,[1] reprints this paper of Professor Osborne Reynolds, and applies it to the rational design of boilers. He also holds the theory that the heat transmitted varies as the velocity of the gases.[2]

High Gas Speeds.—In 1909 Professor J. T. Nicolson carried out to its logical conclusion the application of the Osborne Reynolds theory on a large scale.[3] His paper is well worth a careful study, not only for the results he achieved on the problem of gas speeds, but also for its careful analysis of furnace and chimney losses, and of the most efficient rate of combustion. The substance of what follows, together with Figs. 43–45, have been taken from this paper with the kind permission of the secretary of the Institution of Engineers and Shipbuilders in Scotland.

Dr. Nicolson describes three arrangements which he tried, and he got good results from each. All three supply information which cannot fail to be of use to the designer. Some of this information is negative, but it should be none the less valuable for that.

His main object was to obtain a very high velocity of gas flow along part of the heating surface, but he also embodied other features. One was a second combustion chamber, behind the furnace proper, which he called a reverberatory chamber, and which was intended to complete the combustion of any unburnt gas or particles of fuel which might be drawn off from the fire. Another was to place a counterflow economiser in contact with the gases after they had left the boiler. A third was to use a rotary pump, in addition to the feed pump, to obtain forced or artificial circulation in certain parts of the boiler.

In all three arrangements he used an ordinary Cornish boiler, made by Messrs. Joseph Adamson & Co., of Hyde. This boiler was 24 ft. long and 6 ft. 6 in. diameter. The single Adamson ring flue had an inside diameter of 3 ft. 5 in.

In this flue came first an ordinary grate 6 ft. long with the usual bridge at its back. Then there was a lining of firebrick for a distance of 5 ft 4 in. to form a reverberatory chamber; across the middle of this was built a brick arch to break up the current of hot gases.

The last 10 ft. of the flue was almost completely blocked up by a disbanded steel drum, shown in Fig. 43, 3 ft. 3 in. outside diameter, which therefore left an annular passage only 1 in. wide for the products of combustion. In this way their velocity was greatly increased. This drum had an inner lining which left a space of ½ in. between it and the outer casing. This space was divided up in the form of a quadruple threaded helix, and water was pumped from the bottom of the boiler to the front end of this drum, and forced to flow round these helical spaces, each of which was 9 in. wide, and so encircle the drum with a jacket of water and steam.

[1] "The Steam Engine," John Perry, D.Sc., F.R.S., 1900, p. 594.

[2] *Trans. Inst. Eng. Ship. Scotland*, 1898, vol. xlii.

[3] "Boiler Economics and the Use of High Gas Speeds," *Trans. Inst. Eng. Ship. Scotland*, 1911, vol. liv.

The ordinary feed water after it left the economiser also passed through this water drum, the whole contents of which were discharged from the back end of the drum into the boiler.

Although the objects of this arrangement are not clearly stated in this paper, in part it must have been to accelerate the water circulation. If it was also meant to prevent the drum from overheating, it failed to do so, and in the third arrangement, Fig. 44, it was replaced by a hollow brick plug 3 ft. 2 in. in diameter and 10 ft. long, which therefore left a space of only $1\frac{1}{2}$ in. between it and the inside of the flue.

The first economiser consisted of 163 vertical tubes $\frac{7}{8}$ in. outside diameter, $\frac{3}{4}$ in. bore, and 14 ft. long, placed inside a 16-in. sheet steel

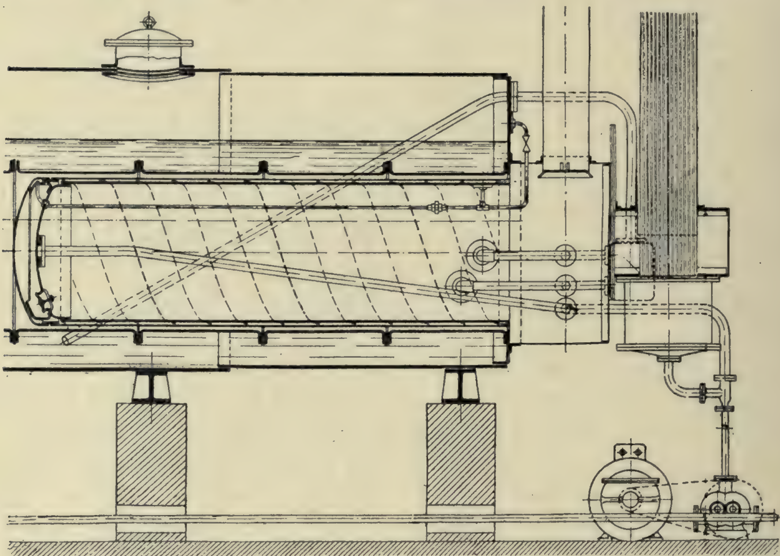


FIG. 43.—Dr. Nicolson's first arrangement.

pipe. To increase the speed of circulation and to force the water against the heating surface, each tube had a $\frac{1}{2}$-in. square bar of iron passed down inside its whole length.

In the first arrangement the feed entered at the top and passed downwards against the upward flow of the flue gases. A 50 h.p. fan was installed to impart the required velocity to the flue gases. This fan required 37·1 B.H.P. when the mean speed of the gases was 101 ft. per second.

It was found that unless the feed pump was kept going at a good speed the feed water remained stagnant in some of the economiser tubes, steam was formed and rose to the top header, where it interfered with the proper working of the counterflow principle.

In the second arrangement the economiser was turned upside down and the flue gases were passed from the top to the bottom.

In the third arrangement, when the water drum was superseded, a second economiser was installed in its place. This was called an

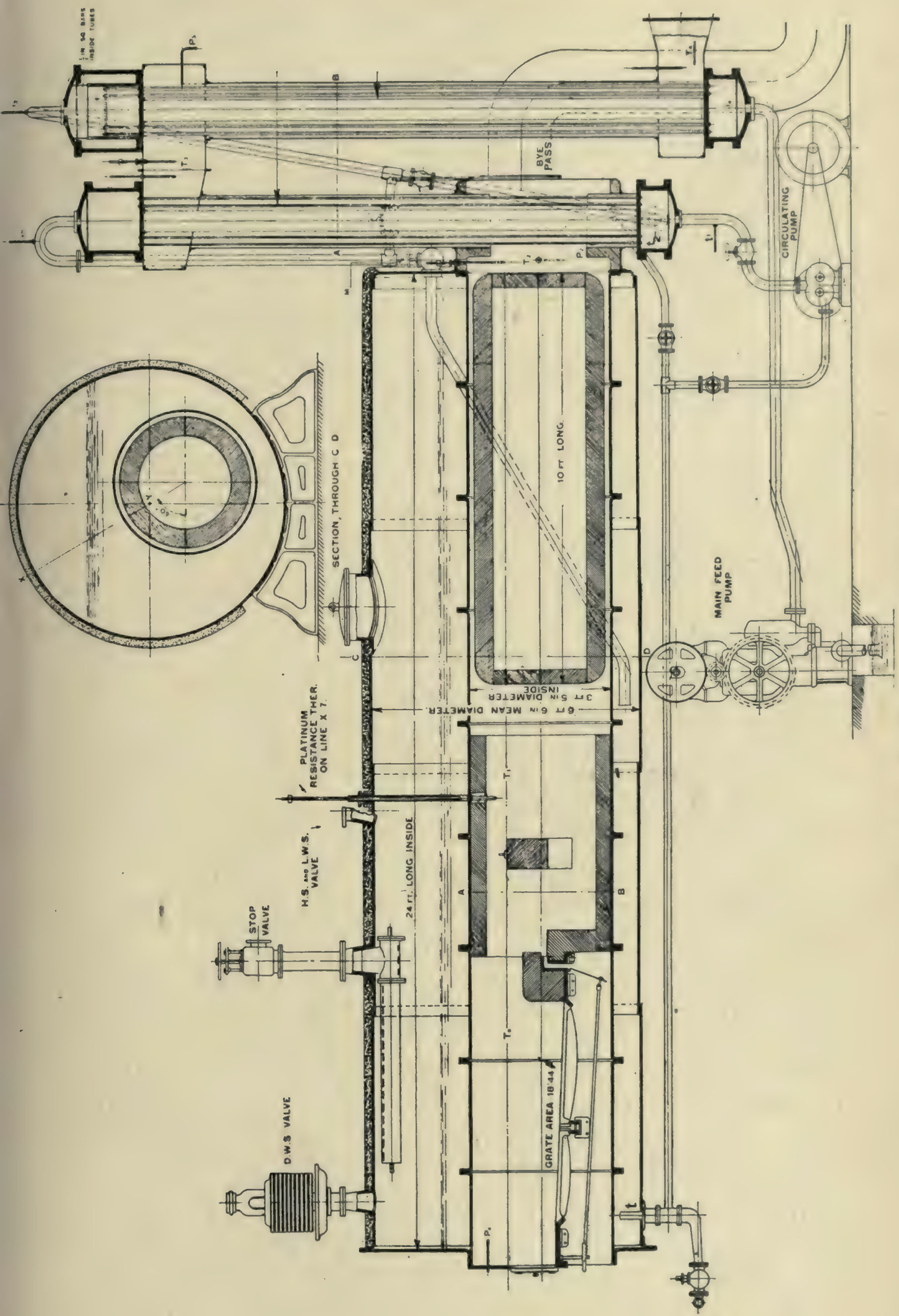


FIG. 44.—Dr. Nicolson's experimental boiler.

evaporator, and water was forced through it as with the drum. The evaporator had 90 tubes, each of $\frac{5}{8}$ in. outside diameter, $\frac{3}{8}$ in. bore, and 12 ft. long. The central portion of the 16-in. containing tube was filled by a 6-in. pipe to which the gas had no access. This evaporator was placed in the flue gases as they flowed upward from the back of the boiler to the top of the original economiser. In the trials that follow it was considered to be an integral part of the boiler.

The feed on leaving the economiser could enter the boiler direct or could be first mixed with the water circulated round the evaporator.

The plant which was tested by Mr. Michael Longridge was practically the same as the third arrangement, but the evaporator had 102 tubes and the economiser 90 tubes, both of the same dimensions as before.

The heating surface of this plant, which is shown in Fig. 44, was as follows:—

Boiler	171	sq. ft.
Evaporator	192	„
Economiser	293	„
Total	656	„

The cross-sectional areas—

For water through the evaporator tubes	0·0782	sq. ft.
„ „ economiser	0·0356	„
For gases. Annular passage, about	1·7	„
„ through the evaporator	0·985	„
„ „ economiser	0·823	„
The grate area, excluding dead plate, was	18·1	„

The first arrangement gave trouble owing to steam forming in the front end and on the top of the water drum, which caused it to get red hot and leak.

With the second arrangement a steam escape pipe was fitted, and although the evaporation was not actually measured, Dr. Nicolson estimated that the average heat transmission in the boiler was about 46,400 B.Th.U. per sq. ft. of heating surface per hour. This must have been obtained with a gas speed of the order of 230 ft. per second. With ordinary present-day boilers the gas speeds vary between 10 and 30 ft. per second, and the heat transmission from 2000 to 8000 B.Th.U. per sq. ft. per hour.

Six independent trials were made by Mr. Michael Longridge, between October 12 and 21, 1909, on the plant shown in Fig. 44.

In his first trial he found that, taking the boiler and evaporator as a unit, 23·06 lb. of steam were evaporated from and at 212° F. per sq. ft. of heating surface per hour. The average gas speed was 105 ft. per second, and the steam pressure 135 lb. per sq. in. absolute. This corresponded to a heat transmission of 18,350 B.Th.U. and required 39·9 lb. of dried fuel per sq. ft. of grate area per hour or $39{\cdot}9 \times \frac{18{\cdot}1}{363}$ $= 1{\cdot}99$ lb. per sq. ft. of heating surface per hour.

If the economiser is included the evaporation from and at 212° F. per sq. ft. of heating surface per hour was 12·75 lb. of steam. For the whole plant the water evaporated per lb. of dried fuel was 11·61 lb. from and at 212° F.

As the calorific value of the dried fuel was 14,929 B.Th.U. per lb. this represented an efficiency of $\frac{11{\cdot}61 \times 970}{14{,}929} = 75$ per cent. Where allowance was made for steam to drive the fan and the circulating pump Mr. Longridge estimated that the net efficiency would be 66·1 per cent.

Dr. Nicolson calculated that the average B.Th.U. transmitted per sq. ft. per hour through the plug flue alone was 32,200 B.Th.U. for this trial.

The lessons to be learnt from these experiments as a whole appear to be that high gas speeds do increase the evaporative power of the heating surface, and that consequently a smaller boiler can be made to do the same duty if weight and space are a consideration. If the counterflow principle is used it is not wise, at any rate with small economiser tubes, to drive the water vertically downwards against an ascending stream of gas. The thermal efficiency, on the other hand, is not improved, though it is well maintained under extremely forced conditions. Dr. Nicolson claimed that the scouring effect of the high speed tends to keep the heating surface thermally clean. The earlier arrangements show that high gas speeds may cause overheating unless high water speeds are also used, and this is a point which is sometimes overlooked.

As a result of his experimental work Dr. Nicolson drew out the design of a high-speed counterflow boiler, which is reproduced in Fig. 45 from the same paper.

The first thing to notice is that although his experiments were done on a Cornish boiler, the design he evolved is a water-tube type. The roof of the furnace even has water tubes instead of the usual firebrick arch. These tubes would receive their heat mainly by radiation. The hot gases pass first into a reverberatory chamber behind the furnace to render combustion more complete, then up through the evaporator shown on the right, across and down through the economiser to the point marked " 7," whence they are then abstracted by a steam turbine-driven fan.

The cold feed is pumped into the drum marked " A " by the feed pump shown at " G." The pipe connections are omitted for the sake of clearness. It rises through the economiser in counter-current to the down-flowing gases to the drum marked " B." It is then led by a pipe to the bottom of the mud drum " C " on the left hand of the drawing, where any sediment is deposited.

The rotary pump marked " F " draws a supply of water from the steam drum "E," and delivers it into a U-shaped channel below the mud drum, whence it is forced along the tubes in the roof of the furnace to the water drum " D," and up back through the evaporator to the steam drum. The clean fresh water supply is collected from the surface of the mud drum and led by a number of bent tubes to the same U-chamber, where it joins the general circulation. The object of the forced circulation is to prevent the evaporator tubes from overheating.

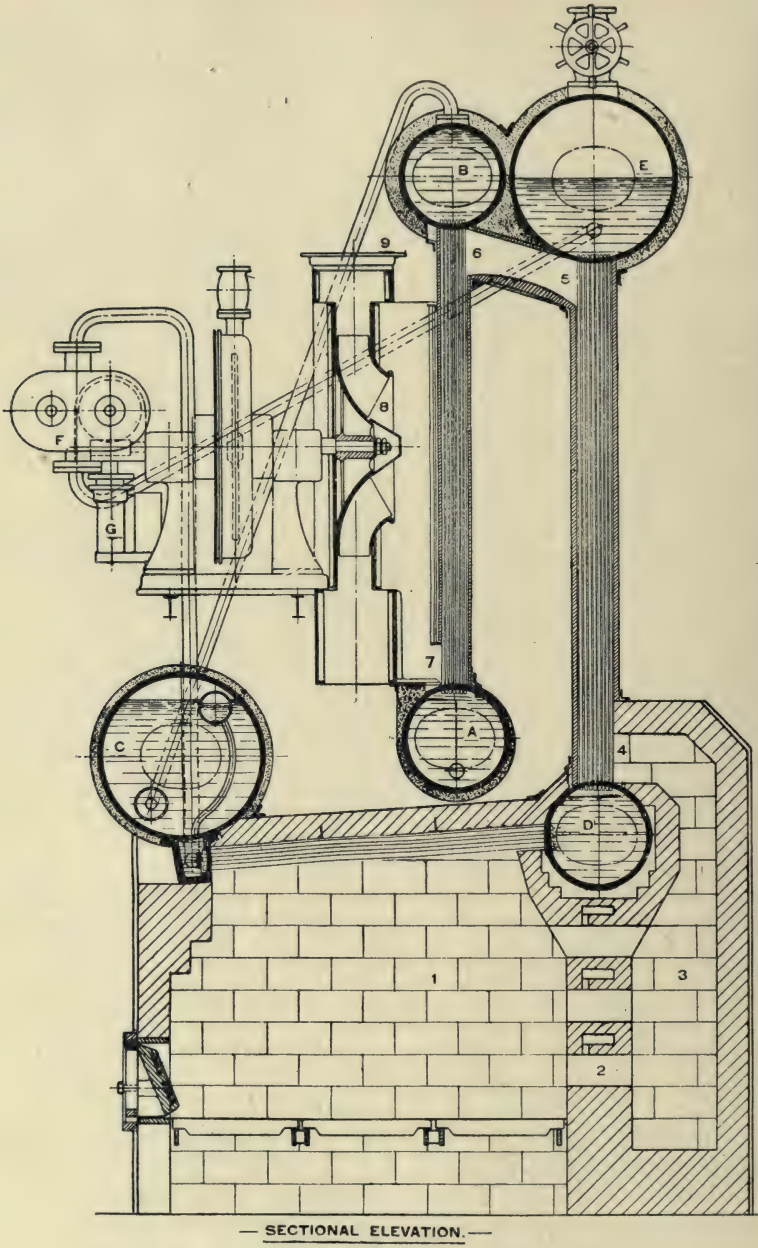


FIG. 45.—Dr. Nicolson's design for a "high-speed" boiler.

Dr. Nicolson points out that the evaporative power of such a boiler can be controlled by the steam supply to the fan turbine, and that, owing to the small volume of water to be heated initially, full steam pressure can be got up from cold within 15 minutes, provided standby steam is available for the turbine.

Although such a design has not been made, it embodies a number of ideas which are worth consideration when a modification of any existing type is contemplated.

The results obtained by Dr. Nicolson have received additional confirmation in a paper by Mr. R. Royds,[1] in which they are compared to a number of other experiments bearing on the same subject. With the exception of the boiler flue with its firebrick plug, when taken alone, they agree very closely with a number of tests carried out in America which, amongst others, are analysed in the paper. As Professor A. L. Mellanby remarks in the discussion, "The actual boiler trials, as analysed by Mr. Royds, are worthy of special study. They show that the Reynolds law was followed in these cases just as it was in the smaller laboratory experiments (a number of which are also described in this paper). They also illustrated very clearly that for a big evaporation per sq. ft. of heating surface high gas speeds were necessary, but for high efficiency the rate of firing must be kept down." He goes on to emphasise a point that was made by Dr. Nicolson, namely that high gas speeds did not necessarily mean high rates of combustion, and that with high draughts the resistance of the heating surface must be so proportioned that only normal quantities of coal would be consumed per sq. ft. of grate.

Other ways of Improving Heat Transmission.—Besides high gas speeds another method for improving heat transmission by conduction which is undoubtedly sound, is to place the heating surface as much as possible at right angles to the flow of hot gases. This improvement may be due to good perforation of the thin film of inert gas which collects on the fire side of boiler plates and tubes, and which keeps off a large amount of heat because of its low conductivity. By drawing the flue gases across instead of along the heating surfaces this inert gas film is probably broken up to a much greater extent at the ordinary gas velocities of 20 or 30 ft. per second. In water-tubes it has the additional advantage of creating by means of suitable baffles a simple means of crossing the water tubes at several different points and so providing a longer path for the flue gases to give up heat. Direct experimental evidence on this point appears to be lacking, but the facts are well recognised by makers of many good types of water-tube boilers.

Since the temperature drop is very much greater between the hot gases and the metal heating surface than between that surface and the water, it should follow that if the fire side of the surface could be increased in area without increasing the water side more heat would be transferred. There are a number of experiments which go to show this is the case.

A good example of its application in practice is the Serve tube. This is an ordinary smoke tube with eight internal ribs or fins running

[1] "Heat Transmission and Efficiency of Boilers," R. Royds, *Trans. Inst. Eng. Ship. Scotland*, 1914–15, vol lviii.

nearly its whole length. An account of comparative trials of marine boilers with and without these tubes was published in *The Engineer*[1] and is abstracted here by the kind permission of the editor.[2]

Two Scotch marine boilers exactly alike, except that one was fitted with Serve tubes and the other with plain tubes, had overall diameters of 10 ft. 6 in. and were 10 ft. 6 in. long. 126 tubes were fitted in both cases, $3\frac{1}{4}$ in. outside diameter and $\frac{1}{8}$ in. thick.

The heating surfaces were as follows:—

Tubes, Serve	1312·9 sq. ft.
„ plain	732·7 „
Furnaces	135·5 „
Combustion chambers	88 „
Total heating surface:—	
With Serve tubes	1536·4 „
With plain tubes	956·2 „
Grate area	31 „

The coal used was Nixon's navigation, which has a net calorific value of about 15,100 B.Th.U.[3]

Both boilers were run at the same time, but provision was made to regulate the draught separately so as to burn the same quantity of coal in each boiler.

The following table gives deductions made from two of the trials reported:—

Trial.	A. 12 hours' duration.		B. 7 hours' duration.	
Boiler with	Serve tubes.	Plain tubes.	Serve tubes.	Plain tubes.
Coal burned—				
Total, in lb.	11,872	11,872	6496	6496
Total, per hour	989·3	989·3	928	928
Per sq. ft. grate area, per hour	31·9	31·9	30	30
Per sq. ft. heating surface, per hour	0·644	1·035	0·604	0·971
Water evaporated—				
Total, in lb.	114,600	103,000	61,500	53,200
Total, per hour	9550	8583	8786	7600
Per lb. of coal as fired, per hour	9·65	8·67	9·47	8·2
Equivalent evaporation from and at 212° F., per lb. of coal	11·19	10·05	11·03	9·50
Per sq. ft. of heating surface, per hour	6·23	8·99	5·73	7·95
Thermal efficiency	$\frac{11{\cdot}19 \times 966}{15{,}100} = 71{\cdot}6\%$	$\frac{10{\cdot}05 \times 966}{15{,}100} = 64{\cdot}3\%$	$\frac{11{\cdot}03 \times 966}{15{,}100} = 70{\cdot}6\%$	$\frac{9{\cdot}50 \times 966}{15{,}100} = 60{\cdot}9\%$

[1] Vol. lxx. (1890), p. 361.

[2] "For the Use of Serve Tubes in Locomotives." See Marchbanks, *Proc. Inst. C.E.*, vol. cxlix. (1902), p. 245.

[3] See Inchley, "Theory of Heat Engines," 1913, p. 243.

It will be seen that with the extra heating surface provided by Serve tubes $11\frac{1}{4}$ per cent. more water was evaporated for the same amount of fuel in Trial A., and $15\frac{1}{2}$ per cent. in Trial B. The table also shows that the water evaporated per sq. ft. of heating surface is not always a true guide to the performance or efficiency of a boiler.

In the trial mentioned on p. 70, the Nicolson boiler and economiser used 7040 lb. of feed water per hour and 737 lb. of coal per hour as fired, so that the water evaporated per lb. of coal as fired was 9·55 lb., a very similar result to this marine boiler of somewhat larger capacity when fitted with Serve tubes. The corresponding equivalent evaporation was 11·37 lb. (or 11·61 lb. for dried fuel, as already given).

In the one case the result was obtained by a very high gas velocity (over 100 ft. per second) and a much smaller heating surface, namely 656 as compared with 1536·4 sq. ft. In the other case normal induced draught was used, but the heating surface was artificially increased on the gas side. If the fins are omitted and the heating surface taken as 956 sq. ft., the results are not so good as a high-speed boiler with 31 per cent. less heating surface. So far as heating surface can be taken as a measure of the weight and size of a boiler this agrees with Nicolson's claim,[1] that for ships there would be a saving of boiler space and weights of 30 per cent.

Radiation.—The transmission of heat by *radiation* takes place only when the heating surface is exposed to the direct action of incandescent gases. Flue gases, unless they are actually glowing, transmit their heat by convection and conduction.

Although radiation is one of the most difficult effects of heat to measure accurately, its importance is becoming more and more realised. Hirsch's experiments, in 1890, on the overheating of boilers, described in Bryan Donkin's book,[2] show that the part of the heating surface directly over the hottest place in the fire evaporated about twice as much water per sq. ft. as the rest of the boiler, and such a result has been confirmed by others.

This points to the increased use of refractory material such as fire-brick placed as near as possible to the heating surface and at the same time played upon by the flame from the furnace. The brick plug in Nicolson's boiler, for instance, was white hot at one end and more than red hot at the other, and consequently the radiation from it was considerable, and probably accounted for his results in the plug flue being greater than the other experiments compared in Royds' paper (p. 73). It will also be noted that he suggests a "radiation evaporator" in the roof of the furnace of his rational boiler.

In the course of some investigations into the mechanism of the combustion of hydrocarbons at low temperatures, which were carried out at Owens College, Manchester, in conjunction with Mr. R. V. Wheeler, Professor W. B. Bone was led to the discovery of what he called surface combustion. As a result of his subsequent researches he established the fact that "all hot surfaces accelerated combustion, and

[1] "Boiler Economics and the Use of High Gas Speeds," *Trans. Inst. Eng. Ship. Scotland*, 1911, vol. liv. p. 146.
[2] "The Heat Efficiency of Steam Boilers," 1898, p. 158.

this acceleration was the greater the higher the temperature, becoming especially marked when the surface was incandescent." [1]

This phenomenon has enabled radiation to be put to an increased and controlled effect in the heat transmission of boilers. With the assistance of the late Mr. McCourt, the Bonecourt gas-fired boiler was evolved, which is described on p. 33. From the test of boiler A on p. 36 it can be seen that the equivalent evaporation per sq. ft. of heating surface reached $\frac{5030}{316} = 15{\cdot}94$ lb. of water, and Professor Bone mentions a mean evaporation of 20 lb. of water per sq. ft. of heating surface, of which 70 per cent. was found to occur in the first third of the length of tubes and 22 per cent. in the next third. Such a steep temperature gradient would tend to keep the water on the move inside the boiler. Another phenomenon was noticed in connection with these tubes. Once the thickness of the scale became about $\frac{1}{32}$ in. any further deposit scaled off and fell to the bottom of the boiler.

The evolution of the Bonecourt boiler is a living example of the development of a practical application from researches in pure science, originally unconnected with the subject.

Water Circulation.—Up to the present time it has been found, in general, that the circulation of the water in a boiler can take care of itself. In drum boilers it is sluggish and usually irregular, in water-tube boilers it is divided by Bertin [2] into three classes.

Limited circulation, in which there is no circulation except that necessary to replace the water as it is evaporated. In this class of boiler all the tubes are connected in series, so to speak, and water enters at one end and leaves as steam at the other. There is no type of this class described in this book, though the French Belville boiler is an example.

Free circulation, in which the tubes are placed more or less horizontally in parallel between two headers which are connected to a steam drum above. The water passes down one header, through the tubes together, and then back as water and steam through the other header to the drum. The Babcock & Wilcox type is an example of this class.

Accelerated circulation, in which a steam drum is connected by a nest of more or less vertical tubes to a water drum below. Larger tubes are sometimes used to return the water from the steam drum to the water drum. Very often there are two or more drums of each class, as in the Stirling boiler.

Messrs. Yarrow & Co., Ltd., have kindly supplied information of some experiments they carried out in 1896 on circulation of this class. They arranged to heat each limb of a long vertical U-tube with a number of Bunsen burners. The tops of the U-tube were connected to a water and steam drum open to the atmosphere. Circulation was started by heating one limb first, and it was found in every case that the current was accelerated in the same direction when heat was applied to the other limb, in which there was a downward flow. It was also noticed that any bubbles of steam generated in the descending

[1] Lecture on "Surface Combustion," before the Cavendish Society of Leeds University, reprinted in *Engineering*, vol. xciii. (1912), p. 632.

[2] "Marine Boilers," 1905, p. 34.

column rapidly disappeared, the steam parting with its heat and raising the temperature of the water. Within the limits of the experiment the speed of circulation depended on the total amount of heat applied, irrespective of how this heat was divided between the down tube and up tube.

Further experiments were carried out under a pressure of 150 lb. per sq. in. in the apparatus shown in Fig. 46. An upward flow was

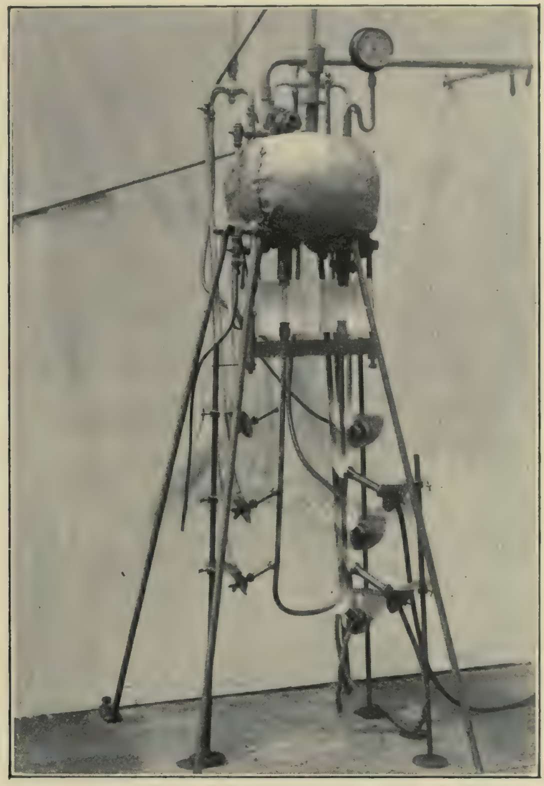


FIG. 46.—Yarrow experiment on accelerated water-circulation.

started in the left-hand limb of the U-tube by lighting the three smaller Bunsen burners. When the five larger burners were lit the circulation was increased and continued *in the same direction* even after the three flames on the up tube were all turned out. It was also found that circulation was more efficiently maintained at a high than at a low pressure, the extreme condition just mentioned of maintaining circulation by applying heat to the down tube only was only established at 50 lb. per sq. in. or over. Between 30 lb. and 50 lb. it was doubtful, and below 30 lb. it was not possible.

Experiments were also made on a full-sized section of a Yarrow water-tube boiler with seven straight tubes working under pressure. At one end there was a furnace under induced draught caused by a steam jet in the funnel. The flames passed transversely across the tubes to this funnel. The circulation was rendered visible by the upper part of the tubes being made of glass. It could clearly be seen that an upward current was established in those tubes nearest the fire, and a downward current in the two or sometimes three tubes furthest away from the fire.

From such experiments it can be seen how automatic and efficient such accelerated circulation is in straight tubes, and how its speed, within limits, adjusts itself to the amount of heat supplied.

Now that transmission between the hot gases and the heating surface is becoming so much more efficient, the question arises whether it will not be necessary to add a fourth class to this division of circulation, namely, *artificial circulation*, where the natural currents of water and steam are augmented and perhaps controlled by some method of pumping. The idea is not new. Professor Nicolson used it in part of his experimental boiler, but probably more with the object of preventing overheating than with any definite idea of increasing the speed of his water circulation as well as his gas flow. In America the oil-fired Talbot boiler, which was described on p. 58, forces water at high velocity through a large number of tubes from top to bottom, and all the steam is formed near the furnace. But here there is no deliberate attempt at high gas speeds. On the other hand, the counterflow principle used is sound.

A long time ago Rankine [1] pointed out that "in a steam boiler, it is favourable to economy of fuel that the motion of the water and steam should on the whole be opposite to that of the flame and hot gas of the furnace. The coolest positions of the water in the boiler should be contiguous to the coolest parts of the furnace and heating surface; and if there is a superheater, it will be most efficient if placed in the hottest part of the furnace."

The future may well be summed up in the words of Professor Perry,[2] "One small tube conveying hot gases, dragged through at an enormous velocity, and a concentric tube conveying water in the opposite direction at great velocity—they had in that combination a method of giving up heat which was fifty times as great as what occurred in an equal amount of heating surface in the best existing boilers."

[1] "A Manual of the Steam Engine," 5th ed. 1870, p. 262.

[2] *Trans. Inst. Eng. Ship. Scotland*, vol. xlii. (1898).

CHAPTER IV

THE CONSTRUCTION OF STEAM BOILERS

To turn now to a few notes on the actual construction of steam boilers. Most boiler manufacturers have a mass of practical experience behind them, which, for the most part, they follow; but a glance at some of the fundamental principles involved may be helpful to the student.

Choice of Type.—Broadly speaking, where floor space is not limited, and where steady steaming is required for an even load, the cylindrical boiler is still considered the most suitable. It is reliable, safe, and has considerable reserve power. With exceptions already referred to, it is not so necessary to use such pure water as the water-tube boilers require.

For high pressures, with fluctuating loads, or where steam is required quickly, it is now customary to install water-tube boilers.

At sea the return multitubular or the water-tube are nearly always used, and the locomotive has developed a type of its own.

Registration Authorities.—In marine work, where reliability and safety are so vital, codes of rules have been drawn up by various authorities, which cover every detail of construction. Before certificates are granted for boilers to go to sea, they have to conform to the rules and pass the survey of the authority concerned.

The principal codes are:—

In The British Empire .	The Board of Trade Survey.
	Lloyd's Register.
	British Corporation.
In France	Bureau Veritas.
In Germany	Germanischer Lloyd's.
In Italy	Registro Italiano Survey.
In U.S.A.	U.S. Board of Supervising Inspectors.

The British Marine Engineering Design and Construction Committee have been co-ordinating these rules. Their work to date is given in a paper by R. E. Seaton, read before the Institution of Naval Architects on April 10, 1919.[1]

The most important codes are those of the Board of Trade and Lloyd's Register. Although they differ in some respects, both sets of rules have proved themselves to be quite safe and reliable. They furnish all the information necessary to construct a marine boiler. They are fully discussed, and many of the formulæ worked out, by C. E. Stromeyer in his book on "Marine Boiler Management and

[1] Reprinted in *Engineering*, vol. cvii. (1919), p. 519.

Construction." [1] Abstracts of them are to be found in most engineering pocket-books.

In Great Britain there is no corresponding authority for land work, but insurance companies, such as the National Boiler and General Insurance Co., Ltd., of Manchester, publish notes on the material, construction, and design of land boilers, which embody sound practical experience.

In the United States the Boiler Code Committee of the American Society of Mechanical Engineers drew up rules [2] for the construction and installation of steam boilers not subject to federal inspection and control (*i.e.* other than marine boilers and boilers of steam locomotives). These rules, which were largely modelled on the Massachusetts Boiler Rules of 1910, have been approved by the American Boiler Manufacturers Association.

The maximum and minimum ultimate tensile strength of mild steel, in tons per sq. in., allowed by some of these authorities is given in the following table:—

Boiler part.	B.T.	L R.	B.V.	A.S.M.E.	National Boiler Ins. Co.
Shell	27-32	28–32	26–32	24½–29	26-30
Flanged ends or flat parts	26–30	26–30	25½–30½	24½–29	24-28
Furnace	26-30	26–30	24–29	24½–28	24–28
Stays and angles . .	27–32	28–32	25½–30½	22·2–28	—
Rivets in bar . . .	26–30	26–30	24–29	21·4–23·2	26–30
Finished rivets . .	23 in shear 27–32 tensile	—	—	—	—
Tubes	23-30	22–25	—	—	—

Wrought iron is not recommended for marine work when it can be avoided. The Board of Trade specify 21 tons with the grain and 17·85 tons across the grain for W.I. shells, and the Bureau Veritas allow the following values:—

Furnace 23 with grain, 21½ across grain.
Rivets and stay bars 23
Tubes 21

Factor of Safety.—For both land and marine boilers the usual factor of safety allowed for boiler shells is 5. It may be as low as 4 in first-class work, or as high as 6 if W.I. is used. For parts subjected to excessive stresses, such as end plates, furnaces, and firebox stays, a safer value to take would be 7–8.

To Obtain the Principal Proportions.—In approaching the construction of any given type of boiler, it is convenient to work backwards from the duty required to the grate area, and then to obtain the heating surface from the grate area according to the ratio selected. Sometimes values of the amount of water evaporated per sq. ft. of heating surface are used to determine the area of the heating surface

[1] Longmans, Green & Co. 5th ed. 1919.
[2] *Trans. Am. Soc. Mech. Eng.* vol. xxxvi. (1914), p. 977.

direct. This is especially the case when non-solid fuels are burnt, but this latter value varies very considerably, both in the different types and in the conditions of draught, etc., under which the boiler may be worked. On the other hand, a fair average figure of 10 lb. of water (from and at 212° F.) per lb. of coal agrees very well for nearly all types of boilers. Verticals are rather less, say 7 to 8 lb., and locomotives are often 12 lb. or more. Oil-fired water-tube boilers reach 15 lb. of water evaporated per lb. of oil.

The coal fired per sq. ft. of grate area can be taken at from 20 to 25 lb., but this figure is more variable, depending as it does on the fuel, the size of the grate, the method of stoking, the conditions of combustion, and the use for which the boiler is required.

All these values represent easy steaming. If the boiler is forced they may be increased 50 per cent. or more.

The quantity of heat that reaches the water has all to pass through the heating surface. Long practical experience shows that there is a ratio of grate area to heating surface area, which gives the most economical results in each type of boiler. Although, of course, this ratio varies in individual examples, for the purposes of preliminary design the following table may be taken as a guide to present-day practice :—

Type of boiler.	Ratio $\frac{\text{heating surface}}{\text{grate area}}$
Vertical	10
,, multitubular	20
Cylindrical (1 or 2 flue)	25–30
Scotch marine (2 flue)	30
Dryback (2 flue)	35
French Elephant	40
Water tube (large tubes)	50
,, (small tubes)	60–75
Locomotive	up to 80

Worked Example of a Lancashire Boiler.—To show more clearly how such principles may be used to obtain the main proportions of a boiler, it might be worth while to work out an actual example of a Lancashire boiler. If a duty of 8000 lb. of steam per hour at a working pressure of 180 lb. per sq. in. is chosen the results may then be conveniently compared with Fig. 15 on p. 27, which is a boiler maker's working drawing of a Lancashire boiler for the same duty and pressure.

Equivalent Duty.—In order to bring the duty of 8000 lb. of steam into line with the above data, it must first be converted into an equivalent evaporation from and at 212° F. To do this it is necessary to know or to assume the temperature of the feed water as it enters the boiler. Suppose this to be 60° F.—

then H at 195 lb. abs. = 1205 (from Callendar's steam tables)

$$t_f = 60° \text{ F.}$$

$$H + 32 - 60 = 1177 \text{ B.Th.U.}$$

$$\text{factor of evaporation} = \tfrac{1177}{970} = 1{\cdot}214$$

$$\text{Equivalent duty} = 8000 \times 1{\cdot}214 = 9712 \text{ lb. per hour.}$$

Grate Area.—Allowing 10 lb. of steam per lb. of coal gives the fuel required as

971 lb. of coal per hour

If $22\frac{1}{2}$ lb. of coal are fired per sq. ft. of grate area the total grate area

$$= \frac{971}{22 \cdot 5} = 43 \text{ sq. ft.}$$

The Lancashire boiler has two flues, so that the area of each grate $= \frac{43}{2} = 21$ sq. ft.

Dimensions of Flue from Grate Area.—The average proportions of Lancashire boilers have been more or less standardised. The furnace and flue tubes are made as large in diameter as possible. Between the two flues and between each flue and the shell requires a minimum gap of from 5 in. to 6 in. For examination and cleaning purposes, about 5 ft. from the back end these flues are reduced about 6 in. in diameter to allow a man to pass from the upper to the lower sides of the flues.

Again, for efficient hand stoking the length of the grate should not exceed 6 ft., otherwise the stoker will have difficulty in spreading his fire evenly right up to the back of the furnace. Applying this length to the area of the grate makes the width of each grate $\frac{21 \cdot 5}{6} = 3$ ft. 7 in. This width practically corresponds with the internal diameter of the flue, which would only be about 1 in. wider. The flue in Fig. 15 is actually 3 ft. $7\frac{3}{4}$ in. internal diameter.

Dimensions of the Boiler Shell from the Heating Surface.—The ratio of the heating surface to the grate area may be obtained from the table on p. 81. For a cylindrical boiler the average value would be $27\frac{1}{2}$.

The area of heating surface required is then $27\frac{1}{2} \times 43 = 1182$ sq. ft.

In a preliminary design the flues may be considered parallel throughout their length. The grate would have a dead plate in front for the first foot of its length and a firebrick bridge at the back, which would also occupy about 1 ft., so that the total length occupied by the grate would be, in this case, 8 ft.

In estimating the heating surface of an internally fired boiler the part of the flue below this length is not counted. The heating surface of the flues exposed to the flame and the hot gases will be, therefore, the internal superficial area of two cylinders, less the area of the half circle below each grate.

It is now necessary to choose an overall length for the boiler, and the following calculations may have to be tried out for one or two values before convenient proportions are obtained. The duty, however, shows that the boiler is one of the largest sizes made in this type, and as Lancashire boilers are made up to 30 ft. in length, this figure will probably be the right one in this case. The heating surface of the flues therefore is arrived at as follows:—

$$2 \times 3 \text{ ft. } 7\tfrac{3}{4} \text{ in.} \times \pi \times 30 \text{ ft.} = 692 \text{ sq. ft.}$$

$$\text{less } 2 \times 3 \text{ ft. } 7\tfrac{3}{4} \text{ in.} \times \frac{\pi}{2} \times 8 \text{ ft.} = 92 \quad ,,$$

$$\text{approximate flue heating surface } 600 \quad ,,$$

This leaves 1182 − 600 = 582 sq. ft. for the shell. The boiler setting on p. 19 shows that the outside of this shell would be swept by the flue gases for two-thirds of its circumference. The area of the exposed segment of the back end less the area of the end of the two flues also forms part of the heating surface. But this may be omitted in a preliminary calculation. The approximate external diameter of the boiler shell (D) may then be found from the relation

$$\tfrac{2}{3}\pi D \times 30 = 582$$

which gives 9 ft. 3 in. for D.

This is near enough to show that the internal diameter would be 9 ft., and the length 30 ft., as is the case with the boiler shown in Fig. 15.

Shell Stresses.—If a boiler drum is considered merely as a thin riveted cylindrical shell, the circular joints are only stressed half as much as the longitudinal seams.[1] The combination of these two principal tensions is shown by Professor J. Goodman[2] to make a boiler shell stand 14 per cent. more pressure before the elastic limit is reached than is given by the ordinary ring theory. This is all in favour of the factor of safety which may be chosen.

The thickness of a boiler shell, therefore, should be calculated from the formula for the stress tending to burst the longitudinal joints—

$$t = \frac{pdx}{2f\eta_p}$$

where t = the thickness of the boiler shell in in.
p = the working gauge pressure in lb. per sq. in.
d = the inside diameter in in.
x = the factor of safety
f = the tensile strength of the boiler plate in lb. per sq. in.
η_p = the plate efficiency allowed for the longitudinal riveted joint.

This formula is the basis of the rules for working pressure allowed by the Board of Trade, Lloyd's, and the American Boiler Code. Lloyd's make an allowance of $\frac{1}{8}$ in. in the thickness for possible corrosion.

For the Lancashire boiler under consideration—

p = 180 lb. per sq. in. (gauge)
d = 108 in.
x = 5
f = 28 tons per sq. in. for landwork (Nat. Ins. Co. table on p. 80)
η_p = 85 per cent. (assumed for a three-row butt-riveted joint)

$$t = \frac{180 \times 108 \times 5}{2 \times 28 \times 2240 \times 85} = 0{\cdot}915 \text{ in.}$$

which is $\frac{15}{16}$ in. (0·9375) to the nearest sixteenth of an in.

Flue Stresses.—The question of flues and other circular parts of boilers which have to withstand external pressure is more complicated. So long as the tube remains circular in section the crushing pressure on

[1] Morley, "Strength of Materials," 1917, p. 313.
[2] Goodman, "Mechanics Applied to Engineering," 9th ed. 1918, p. 402.

the circumference may be found with a formula of the same kind as the bursting stress of the shell, namely—

$$t = \frac{pd}{2f}$$

introducing a factor of safety—

$$t = \frac{pdx}{2f} \text{ for welded or for seamless tubes} \quad . \quad . \quad . \quad (a)$$

$$= \frac{pdx}{2f\eta_p} \text{ for longitudinal riveted flues} \quad . \quad . \quad . \quad . \quad (b)$$

Sir W. Fairbairn[1] carried out a number of experiments under water pressure on thin tubes made of tin, and found that the length of the tube was a factor in the collapsing pressure. This was due to the fact that if the tubes are long the material buckles or collapses before the crushing stress, as given by the above formula, is reached.

The formula deduced by Sir W. Fairbairn was of the following kind:—

$$\text{The collapsing pressure } (p) = \frac{Ct^{2\cdot19}}{ld}$$

where l is the length of the tube, and C is a constant which he determined by his experiments. For practical purposes it has been found accurate enough to take the square of the thickness, and in this form is used by the Board of Trade and Lloyd's.

Professor W. C. Unwin[2] analyses these results, and shows that there is a clear rule connecting the number of lobes into which the tube collapses with the ratio of length to diameter of the tube. His work should be referred to if accurate determinations are required, connecting the length and the thickness with the diameter.

To bring the strength of flues well within the crushing formula and to avoid all fear of collapse taking place first, a number of corrugated or ribbed flues have been designed and patented.

For such furnaces formula (a) (p. 84) may safely be applied. Those used by the Board of Trade and Lloyd's are based on this formula, in which constants have been obtained as a result of careful experiments on full-size flues under water pressure. For a description of these tests, as well as of several patented furnaces, a very practical paper, by D. B. Morison, on "Marine Boiler Furnaces"[3] should be consulted.

In the Adamson boiler on p. 27 (Fig. 15)—

p = 180 lb. per sq. in.
d = 3 ft. 9 external diameter
x = 8 for furnaces
f = 26 tons per sq. in. (from National Boiler Code in table on p. 80)

$$t = \frac{180 \times 45 \times 8}{2 \times 26 \times 2240} = 0\cdot56 \left(\tfrac{9}{16}\right) \text{ in.}$$

It actually is $\frac{5}{8}$ in. (0·625), which shows that the factor of safety is nearer 9 than 8 in this case.

[1] *Proc. Royal Society*, 1858, p. 234.
[2] Unwin, "Machine Design," Part I. 1916, p. 113.
[3] Published in *Proc. N.E. Coast Inst. Eng. and Shipbuilders*, vol. ix. (1892).

D. B. Morison points out on p. 127 of his Paper that low tensile steels for furnaces have their good points, as they are generally more ductile and less liable to crack, and it is therefore also possible that the makers have used steel of a lower value than 26 tons per sq. in.

Stresses in Ends and Flat Surfaces.—The experimental theory of flat plates was originally worked out by Grashof, and is thoroughly treated by C. Bach,[1] but, as Professor W. C. Unwin[2] points out, assumptions have to be made as to the dangerous section and the position of the reactions of the supports. The most recent theoretical treatment of the subject is given by A. Morley.[3]

Flat plates under pressure are subjected to bending and to buckling stresses. Theory shows that the load they can withstand varies as the square of the thickness and inversely as the supporting area. In this form, with suitable constants and allowances, the formula appears in all boiler rules. The flat plate requires staying or supporting, if it forms part of the heating surface. This is always done on the water side of the surface, as any projections exposed to the temperature of the fire or the hot gases get burnt away and depreciate very rapidly. To calculate the least proportion of these supports, the surface area is divided between each gusset or stay, and their strength, and therefore dimensions, computed from the total pressure on such areas.

This is the method adopted by Unwin,[4] who shows that the least area (a) of such a support is given by the formula—

$$p\frac{Ax}{f} \times \frac{l}{y}$$

where p = pressure in lbs. per sq. in. (gauge)
A = area of flat plate assigned to the stay in sq. in.
x = factor of safety
f = tensile strength of stay in lb. per sq. in.
y = mean horizontal distance between the two joints of the stay
l = actual mean distance between the two joints of the stay.

In practice such stays, particularly if they are gusset stays, are often made considerably stronger than necessary for tension. This is done for the sake of stiffness.

Consider, for instance, the three gussets in Fig. 15 marked H, I, and J. The area A which each supports would be for H about 840 sq. in., for I about 726 sq. in., and for J about 432 sq. in.; $p = 180$ lb. per sq. in., $x = 5$, and f may be chosen at 25 tons per sq. in. The ratio $\frac{l}{y} = \frac{6}{5}$. Hence the areas ($a$) at their weakest section—

$$\begin{aligned} \text{for H} \quad a &= \frac{180 \times 840 \times 5}{25 \times 2240} \times \frac{6}{5} \\ &= 0{\cdot}01925 \times 840 = 16{\cdot}2 \text{ sq. in.} \\ \text{for I} \quad a &= 0{\cdot}01925 \times 726 = 14{\cdot}0 \quad ,, \\ \text{for J} \quad a &= 0{\cdot}01925 \times 432 = 8{\cdot}3 \quad ,, \end{aligned}$$

[1] Bach, "Elastizität und Festigkeit," 6th ed. 1911, pp. 535 *et seq.*
[2] Unwin, "Machine Design," Part I. 1916, p. 128.
[3] Morley, "Strength of Materials," 1917, pp. 422 *et seq.*
[4] Unwin, "Machine Design," Part I. 1916, p. 171.

The breadth of the gusset may be measured at the first rivet of the horizontal joint, at right angles to l.

For H $b = 36$ in. as made
for I $b = 30$ in. ,,
for J $b = 15$ in. ,,

Hence the gussets need only have the following thicknesses :—

$$H = \frac{16 \cdot 2}{36} = 0 \cdot 45 \text{ in.}$$

$$I = \frac{14 \cdot 0}{30} = 0 \cdot 47 \text{ in.}$$

$$J = \frac{8 \cdot 3}{15} = 0 \cdot 55 \text{ in.}$$

whereas they are all made of $\frac{13}{16}$ in. ($0 \cdot 8125$) boiler plate.

The diagonal stay rod in place of a gusset stay is not common in English practice, though frequently used abroad. The American H.R.T. boiler in Fig. 2, p. 13 shows diagonal stay rods $2\frac{1}{2}$ in. diameter. The central stay here supports an area of about 270 sq. in., and has a ratio $\frac{l}{y} = \frac{11}{10}$. Applying the formula just used would give a least cross-sectional area of

$$\frac{185 \times 270 \times 5}{25 \times 2240} \times \frac{11}{10} = 4 \cdot 9 \text{ sq. in.}$$

which corresponds to $2\frac{1}{2}$ in. diameter.

If the ends of the drums are made dish ended and flanged, they are so much strengthened in themselves that gussets or supports are not needed. This method is often used for water-tube drums and is one of the special features of the Lancashire type of boiler made by Messrs. John Thompson & Co., Ltd., of Wolverhampton, who have kindly supplied Fig. 47. Both ends of this boiler are dished and increased in thickness to give the same factor of safety as the rest of the shell. The only rivets in these ends are round the flanged circumference and at the mouth of the flues, where they are protected by firebrick about the grate. In both cases these rivets are subject to shear instead of to tension, as would be the case with gusset stays. Since the ends are rigid the flue has corrugations, of the "Fox" type in this case. The opening in the front end of the boiler is large enough to allow the flue to be withdrawn and refixed without dismantling the end plate.

Fig. 48 shows the process of dish ending and flanging the drum end of a Yarrow water-tube boiler. The circular plate is heated in the furnace at the back and passed down rollers and guide rails to the hydraulic press in the foreground. The stamp descends into a corresponding die underneath and the plate is dished and flanged in one operation.

Plates that are weakened by the insertion of nests of tubes require to be considerably thicker. This can be seen in Fig. 49, which shows a Yarrow steam drum plate being drilled with six holes at once. A jig plate with accurately spaced holes, guides the twist drills and ensures clean-cut holes at the correct angle.

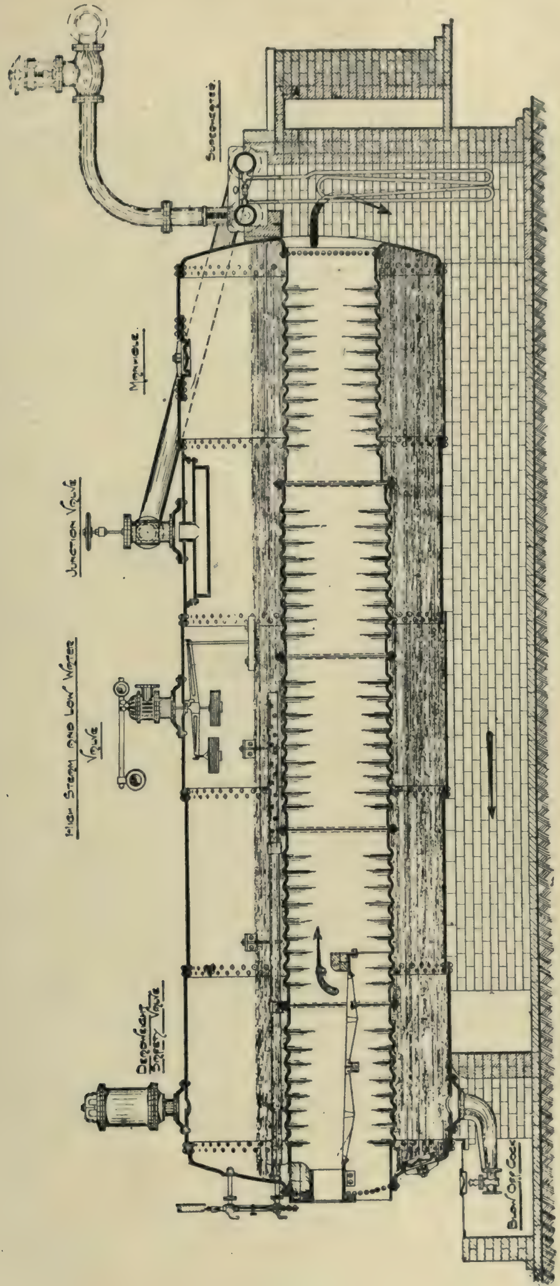


FIG. 47.—Thompson's dish-ended Lancashire boiler.

When such a boiler is being assembled, the tubes are driven through these holes and expanded with a rotary roller expander which also slightly opens out their ends. This operation can be seen in Fig. 50.

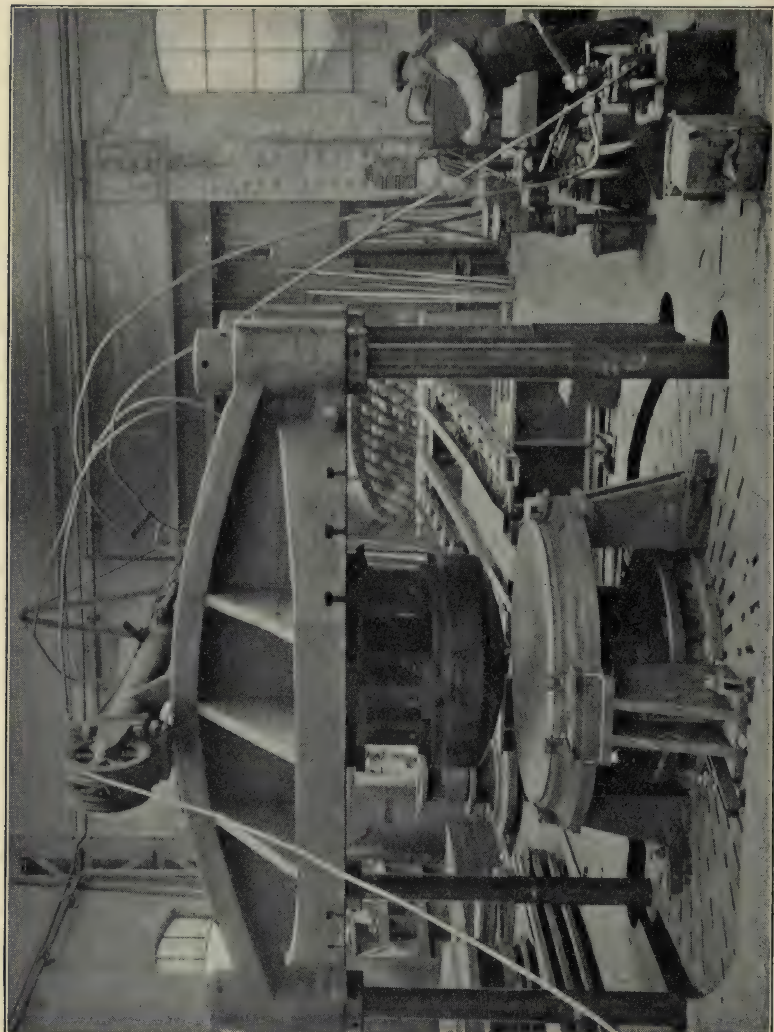


FIG. 48.—Dish-ending and flanging.

Riveted Joints.—The principles underlying the design and construction of riveted joints are very clearly set out by Professor W. C. Unwin[1] or by Professor J. Goodman.[2]

The boiler codes and registration authorities cover in detail nearly

[1] Unwin, "Machine Design," Part I. 1916, chap. iv.

[2] Goodman, "Mechanics Applied to Engineering," 9th ed. 1918, pp. 407 *et seq.*

all the types of actual joints used in modern practice and give formulæ for their design.

The ratio which the strength of a unit strip of the riveted joint bears to the same length of solid plate is known as the efficiency of the joint.

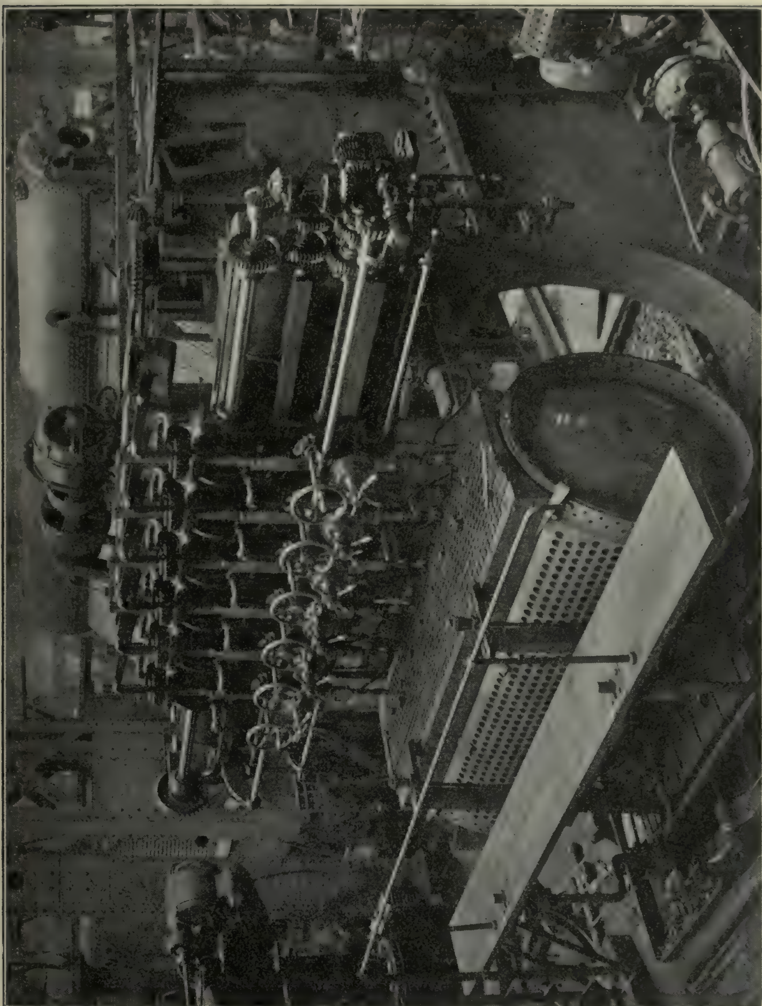


FIG. 49.—Drilling six water-tube holes at once.

This efficiency is usually calculated on the strength of the plate as weakened by the rivet holes across the outer or largest pitched row. If P = this pitch in in. and d the diameter of the rivet hole in in.

$$\text{the plate efficiency } \eta_p = \frac{P - d}{P}$$

The efficiency can also be calculated on the resistance to shear of

the number of rivets in a unit strip of the joint, compared to the tensile strength of the solid plate.

If n = number of rivets per unit strip
a = cross-sectional area of the rivet in sq. in.
P = maximum pitch in in. of the rivet, usually the width of the unit strip
t = the thickness of the plate in in.
f_t = the tensile strength of the plate in tons or lb. per sq. in.
f_s = the shear strength of the rivet material in corresponding tons or lb. per sq. in.

$$\text{then the rivet efficiency } \eta_r = \frac{f_s \times n \times a}{f_t \times P \times t}$$

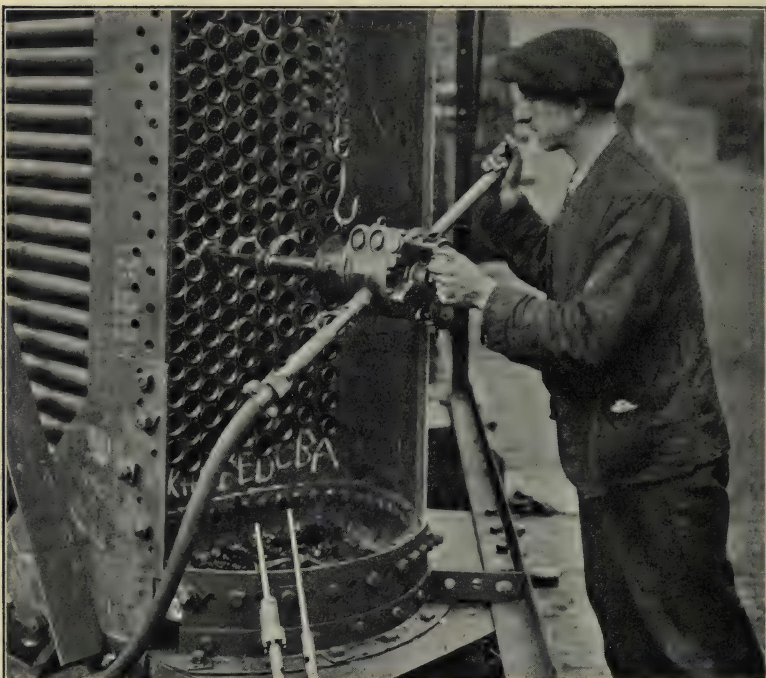


FIG. 50.—Expanding water-tubes.

It is customary when designing joints so as to make these two efficiencies equal, to take $2a$ for all the rivets in double shear, but the Board of Trade only allow $1{\cdot}875a$ and Lloyd's $1{\cdot}75a$ for double shear. J. Goodman[1] shows that the ratio $\frac{f_s}{f_t}$ is 0·80 when Poisson's ratio is 4.

In order to get as high a plate efficiency as possible multiple-riveted joints are so designed that the outer row has a larger pitch than the

[1] Goodman, "Mechanics Applied to Engineering," 9th ed, 1918, p. 400.

others. There is a practical limit to this maximum width beyond which the plate will not caulk satisfactorily. J. Goodman [1] gives this pitch as six times the diameter of the rivet. The Board of Trade fixes a maximum pitch of $10\frac{1}{2}$ in. (formerly 10 in.). The longitudinal joint shown on the American H.R.T. boiler in Fig. 2 is made with the outer butt strap saw toothed, so as to get an increased efficiency without preventing satisfactory caulking. In this joint the diameter of the rivet is $\frac{7}{8}$ in., but the pitch is 12 in. or 13·7 times the rivet diameter, whilst the plate efficiency

$$\eta_p = \frac{12 - \frac{7}{8}}{12} = \frac{89}{96} = 92·7 \text{ per cent.}$$

The operation of caulking the circular seam of a Yarrow steam drum can be seen in Fig. 51. It will be noted that this drum has a double zigzag lap-riveted circumferential joint and a double zigzag butt-riveted longitudinal joint with butt straps or cover plates. The water drums have a single row of countersunk rivets and the same longitudinal seams as the steam drum. These countersunk rivets are hand riveted as shown in Fig. 52. Fig. 51 also shows in the foreground a modern man-hole lid which fits on to the flanged hole of the boiler end lying on the ground. Another one of these lids is shown in position on the water drum. The extra thickness of the plate where tubes are fitted is also shown very clearly.

J. Goodman [2] points out that the bearing pressure on a boiler rivet must not exceed 50 tons per sq. in. on the area measured by the product of the rivet length and its diameter. It is better to keep this value below 45 tons per sq. in. This pressure, however, is seldom reached in modern design and is often not taken into account for that reason.

There is a tendency to use smaller-diameter rivets than formerly. Unwin [3] gives

$$d = 1·2\sqrt{t} \text{ to } 1·4\sqrt{t}$$

Goodman's [4] tables are based on

$$d = 1·1\sqrt{t}$$

It should, however, not be forgotten that the plate efficiency formula more nearly represents the actual condition of affairs than that for rivet efficiency. The plate as weakened by the outer row of rivet holes would tear more or less along that row, whereas all the rivets on one side of the unit strip would have to shear before the joint failed in that way, and the rivet shear formula takes no account of the frictional resistance between the plate surfaces, which may be considerable but which is difficult to estimate. Experience has shown that smaller diameters may be used than would appear safe from such theoretical considerations.

The type of joint to use in the various parts of a boiler is largely left to the experience of the designer. It will be noted in the

[1] Goodman, "Mechanics Applied to Engineering," 9th ed. 1918, p. 415.
[2] *Ibid.* 9th ed. 1918, p. 415.
[3] Unwin, "Machine Design," Part I. 1916, p. 136.
[4] Goodman, "Mechanics Applied to Engineering," 9th ed. 1918, p. 409.

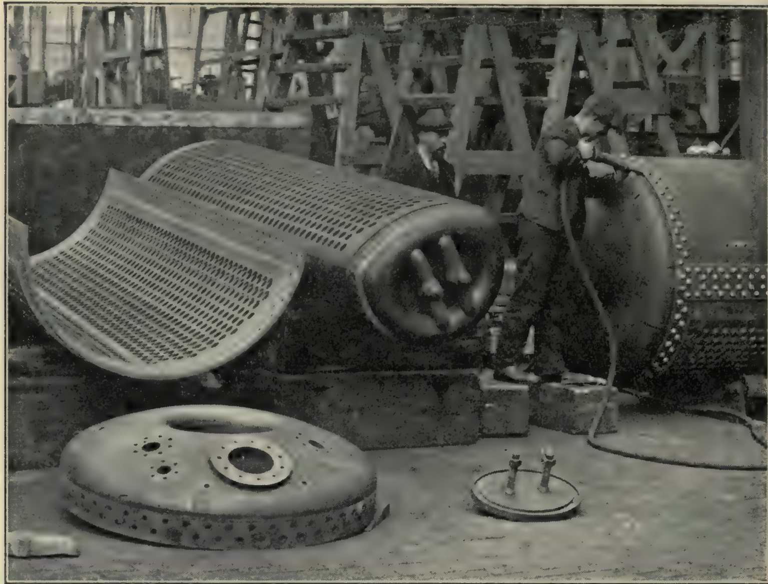


FIG. 51 —Caulking.

FIG. 52.—Hand-riveting.

Lancashire boiler, in Fig. 15, that the circular seams are double zigzag lap riveted except at the front and back ends, where larger single-row rivets are used. This longitudinal seam is common in British practice. It consists of a treble-riveted butt strap joint, but the two straps are of different widths and thicknesses. The inner one, covering three rows of rivets, is $\frac{1}{16}$ in. thinner than the outer one, which covers only two rows of rivets, and the outer row of rivets is larger than the others.

In approaching the design of any particular joint it is a good plan to work out first the maximum pitch that is required for equal tearing of the plate and shearing of the rivets in a unit strip, that is a strip in which the arrangement of the rivets repeats itself. In the absence of any table the relation between the thickness of the plate (t) and the diameter of the rivet hole (d) may be taken as

$$d = 1{\cdot}1\sqrt{t} \text{ to the nearest } \tfrac{1}{16} \text{ of an in.}$$

e.g. for a $\frac{15}{16}$ in. plate

$$d = 1{\cdot}03 = 1 \text{ in. diameter.}$$

This pitch may then be checked for caulking or for minimum distance, namely, not less than $2d$ to enable the rivet head to be made properly, and in the more complicated joints for bearing pressure on the rivets. Lastly, if an efficiency of the joint has been assumed in arriving at the thickness of the boiler shell the pitch should be compared with that given by the formula—

$$\frac{P - d}{P} = \eta_p$$

A consideration of the pitches obtained by these various methods forms a guide to the actual pitch decided upon, which must, of course, fit in evenly with the total length of the joint.

In joints such as the central ring joints or longitudinal joints, and some of the gussets which are called upon to resist the load of the boiler without assistance, it is wise to work out the minimum number of rivets required to take the total load on the joint. Such a calculation requires the assumption of a factor of safety, which may be taken as 5 in the absence of specific information as to the safe load which the rivet material will hold.

SECTION II

STEAM PRIME MOVERS

CHAPTER V

GENERAL CONSIDERATIONS AND THERMAL EFFICIENCY

The use of steam as a medium to obtain power dates back at least to the beginnings of the eighteenth century, when Savery introduced his pumping engine in 1698, and Newcomen developed his atmospheric engine in 1705. James Watt took out his first patent in 1769 and lived till 1819. The first quarter of the nineteenth century saw the application of steam to locomotives by Trevithick and George Stephenson, and to ships by Symington, Bell, and Fulton. The last quarter saw its application to steam turbines by the Hon. Sir Charles Parsons. It is safe to say that the work of these pioneers has contributed more than any other single engineering conception to the vast changes in the lives of civilised peoples which have taken place in the last hundred years.

The first natural idea that would occur to any one, on seeing steam issuing from a nozzle, would be to utilise its kinetic energy directly by means of vanes or against the air. This was actually done by the ancients in classical times, though their apparatus was little more than a toy.

The second method, which would result after further study of the peculiar properties of steam, would be to allow the steam to expand in a closed vessel against a moving piston. It was along these lines that James Watt developed the reciprocating steam engine with its separate condenser, to such a stage that little remained for his successors to do except to perfect his ideas. Early pioneers were, of course, hampered in their designs by the lack of high tensile steels and specialised alloys, but their workmanship was so good that they achieved results which were little short of wonderful.

The accumulating experience of nearly a hundred years had the effect of establishing the reciprocating steam engine on a very firm basis, and it was only when the theoretical study of the conditions under which it was constrained to work appeared to show that there was little room for improvement in economy, that the rotary idea began to be seriously considered again. In 1884, as the result of numerous experiments and a careful study of the underlying principles, Sir Charles Parsons constructed his first practical turbine, and now the steam

turbine more than holds its own with the piston engine, particularly when large power units or economy in floor space are required.

Thermal Considerations—Rankine Efficiency.—Although nowadays there are two distinct methods of utilising steam to obtain power, either by using its expansive property behind a piston, or by the conversion of some of its kinetic energy into work as it impinges or reacts on moving blades, in both cases the energy first exists in the steam in the form of heat, and the number of heat units converted into work forms a means of measuring the thermal efficiency of either a steam engine or a steam turbine.

Various methods have been proposed as a basis for comparison, depending on the cycle of operations that the steam undergoes in an ideal case, where radiation and conduction heat losses are assumed to be absent. Such methods may be studied in books on Thermodynamics.[1] A committee on the thermal efficiency of steam engines appointed by the Institution of Civil Engineers [2] recommended that the Rankine (or Clausius [3]) cycle be adopted as a basis of comparison, and as this standard is now common a few considerations of how this may be conveniently done in practice may be of use.

The Rankine cycle represents the heat requirements of an ideal heat engine using steam with adiabatic expansion as a working fluid between any given temperatures.

It may be calculated from the following two formulæ, which represent the number of heat units per lb. of steam used :—

$$(T_1 - T_2)\left(1 + \frac{x_1 L_1}{T_1}\right) - T_2 \log_\epsilon \frac{T_1}{T_2} \quad . \quad . \quad . \quad . \quad (1)$$

for saturated steam; or

$$(T_1 - T_2)\left(1 + \frac{L_1}{T_1}\right) + C_p (T_s - T_1) - T_2\left(\log_\epsilon \frac{T_1}{T_2} + C_p \log_\epsilon \frac{T_s}{T_1}\right) \quad (2)$$

for superheated steam.

In these formulæ—

T_1 = the absolute temperature of the saturated steam as supplied to the heat engine.

T_2 = the absolute temperature of the working fluid rejected by the heat engine.

T_s = the absolute temperature of the steam if superheated.

(In all three cases these values represent Fahrenheit scale + 460°, or Centigrade scale + 273°.)

L_1 = the latent heat of the steam corresponding to T_1.

x_1 = the dryness fraction of the steam at T_1.

C_p = the mean specific heat of the superheated steam at constant pressure, which may be obtained direct from the curves in Fig. 1 (p. 5).

$\log_\epsilon = 2{\cdot}3 \log_{10}$.

[1] Such as Inchley's "Theory of Heat Engines," 2nd ed. 1920, or Ewing's "Thermodynamics for Engineers," 1st ed. 1920.

[2] *Proc. Inst. C.E.* (1902) vol. cl. p. 218; see also *ibid.* (1913) vol. cxcv. p. 265.

[3] Ripper's, "Steam Engine Theory and Practice," 7th ed. 1914, gives on p. 59 the differences between the Rankine and the Clausius cycle.

The ideal thermal efficiency, often referred to as the Rankine cycle efficiency, is obtained by dividing these two formulæ by the total heat available per lb. of steam in each case, namely, formula (1) by

$$x_1 L_1 + T_1 - T_2$$

for saturated steam ; or formula (2) by

$$L_1 + T_1 - T_2 + C_p(T_s - T_1)$$

for superheated steam.

Charts in several forms[1] have been prepared to show, among other properties of steam, formulæ (1) and (2) worked out for dry or wet saturated steam, and also to include superheat. One of the most convenient is the Mollier diagram, which is given in any modern book of steam tables, and shows total heat units (B.Th.U. per lb. of steam) plotted against entropy. In one form this chart has constant pressure curves running diagonally across one way, and the conditions of superheat or wetness shown by curves running diagonally the other way.

To find the solution of formulæ (1) or (2), that is the number of B.Th.U. per lb. of steam between any two given temperatures according to the Rankine cycle, find the corresponding pressures in lb. per sq. in. absolute from steam tables, note the B.Th.U. corresponding to the point of intersection of the higher pressure with the quality line of the steam, then note where the same entropy line cuts the lower pressure curve and read off the B.Th.U. again. The difference between these two readings will be the adiabatic heat drop or the number of B.Th.U., according to the Rankine cycle. This follows from the fact that lines of constant entropy represent adiabatic expansion of the steam.

Occasionally, particularly in steam turbine work, this heat drop is required more accurately than can be obtained from a Mollier diagram, unless the chart is on a very large scale.

Heat drop tables, or temperature-entropy tables, may then be used. The most authentic results so far obtained in the calculation of the adiabatic properties of steam are due to Professor H. L. Callendar, whose work is the only one that is consistent throughout.[2] H. Moss has calculated the adiabatic heat drop per 1 lb. of steam from Callendar's work and embodied the results in heat drop tables,[3] which range from 400 lb. sq. in. pressure to 50 lb. sq. in. pressure and under, and give the heat drop direct for every 25° F. of superheat from 0° F. to 300° F., corresponding to all vacuums from 27·0 in. to 29·1 in. in increments of 0·1 in. These tables, which can be obtained either for gauge pressures or for absolute pressures, were published for the turbine section of the British Electrical and Allied Manufacturers Association (BEAMA), and intermediate values may be quickly calculated by the use of approximate interpolation. They are correct to 0·01 B.Th.U.

[1] See, for instance, a steam alignment chart by D. H. Thomson, which gives any four properties of steam when two are known, *Engineering*, vol. cix. (1920), p. 301.

[2] See H. L. Callendar, "The Properties of Steam," 1920.

[3] "Heat Drop Tables," 1917, published by Edward Arnold.

Another form of table which supplies other information besides the heat drop may be found in Peabody's steam and entropy tables,[1] and is known as a temperature-entropy table. These tables give the pressure in lb. per sq. in. absolute, the quality, the total heat contents per 1 lb. of steam in B.Th.U., and the specific volume in cubic feet per lb. for each degree Fahrenheit from 420° to 85°, and for each hundredth of a unit of entropy between 1·52 and 1·83. A broken line corresponds approximately to the dry saturated line of the Mollier chart. The figures in the quality column to the right or above this line, are degrees of superheat, to the left or below they represent the dryness fraction (x_1) to four places of decimals with the decimal point omitted. Intermediate points may conveniently be obtained by interpolation on the assumption that the curve between two adjacent values is a straight line.

The temperature-entropy tables are only worked out to 0·594 lb. per sq. in., but here, again, it is safe to extrapolate down to 0·5 lb. per sq. in. to obtain the B.Th.U., or specific volume.

It should be noted that Peabody's tables are based on Knoblauch and Jakob's values for the specific heat of superheated steam, and are therefore not so accurate as Callendar's steam tables; but they are the only tables so far which give the temperature-entropy relation in sufficient detail to be of value in estimating the heat drop per stage of a steam turbine.

Ideal Steam Consumptions.—The steam consumption of an ideal engine or turbine may be conveniently plotted on such a diagram as Fig. 53. The number of B.Th.U. corresponding to one horse-power per hour would be

$$\frac{33{,}000}{778} \times 60 = 2546 \text{ B.Th.U.}$$

or to one Kilowatt per hour—

$$2546 \times \frac{1000}{746} = 3412 \text{ B.Th.U.}$$

These values, divided by the number of B.Th.U. in the Rankine cycle corresponding to the given conditions under which any particular engine is constrained to work, give the number of lb. of steam required by the ideal engine per H.P. or per K.W., as the case may be.[2]

Coefficient of Performance or Efficiency Ratio.—It is becoming more and more recognized that the rational basis of thermal comparison between one steam engine and another is to compare its actual performance with the ideal comparison outlined above. Such a factor supplies the designer with information as to how near he is approaching the ideal result possible with his engine, and by estimating the

[1] "Tables of the Properties of Steam," C. H. Peabody, 8th ed. 1914, published by John Wiley & Sons.

[2] For curves giving the number of B.Th.U. per minute per H.P. and the number of calories per minute per K.W., see *Proc. Inst. C.E.* vol. cxcv. (1913–14), pp. 362 and 363.

necessary heat losses due to friction, radiation and conduction, etc., it is possible to arrive at the margin of improvement still available. This factor, which is known as the coefficient of performance, or as the efficiency ratio, is obtained from the ratio

$$\frac{\text{ideal result}}{\text{actual result}}$$

and may be calculated either on heat units or consumptions. It

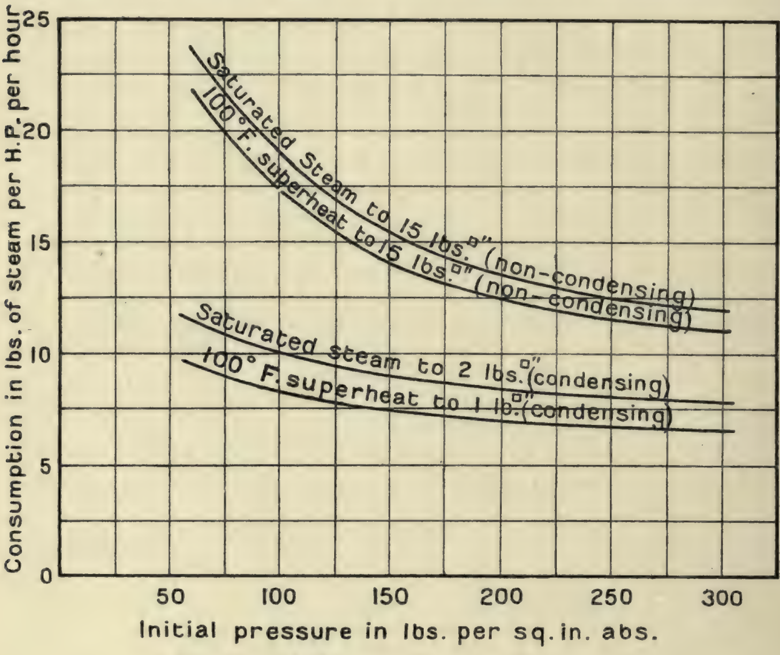


FIG. 53.—Ideal consumption curves for steam.

should not be confused with the thermal efficiency of the engine which is often recorded in trials and which represents the number of heat units turned into work divided by the total number of heat units available in the steam (p. 96).[1]

Steam Turbine Tests.—It is at present customary to consider the efficiency of a steam turbine for test purposes as

$$\frac{\text{actual work done on turbine shaft}}{\text{adiabatic heat drop} \times 778}$$

but as an increase in either superheat, pressure, or vacuum reduces the steam consumption and *vice versâ*, standard steam conditions are taken,

[1] For complete test reports of Steam Engines and Boilers, see *Proc. Inst. C.E.* vol. cxcv. (1913–14), pp. 265–367. See also Royd's "The Testing of Motive Power Engines" (1911), chaps. v., vi., vii., and viii., and Inchley "The Theory of Heat Engines" (2nd ed. 1920), chap. xvii.

to which the performance of all turbines may be reduced by applying proper corrections.

These conditions, as first suggested by K. Baumann,[1] are as follows:—

For high-pressure turbines—

180 lb. per sq. in. gauge and 150° F. superheat both measured before the turbine stop.

28 in. vacuum, measured at the turbine exhaust flange.

For low-pressure turbines—

16 lb. per sq. in. gauge measured before the turbine stop valve 0° F. superheat.

27½ in. vacuum, measured at the turbine exhaust flange.

Barometer 30 in. in both cases.

Correction Tables[2] have been calculated by C. H. Naylor for the turbine section of the BEAMA, which are based on the correction curves for pressure, superheat, and vacuum given by Baumann in the above paper. These tables cover a range of pressure from 50 lb. to 300 lb. per sq. in. gauge in increments of 10 lb., and a vacuum from 27·0 in. to 29·1 in., decreasing by 0·1 in. at a time. The superheat rises from 0° F. to 300° F. in steps of 25°, and there is also a low-pressure table showing terminal pressures and corresponding vacua, for initial pressures from 14 to 18 lb. per sq. in. absolute, rising 1 lb. at a time. The equivalent standard efficiency corresponds to the actual efficiency obtained multiplied direct by the correction factor given in these tables.

Definitions. Steam Turbines.—All steam turbines receive their steam at a higher pressure, extract work from it by passing the steam through various arrangements of nozzles (or fixed blading) and moving blades, and reject the steam at a lower pressure and consequently at a larger volume.

If all, or nearly all, this drop in steam pressure takes place in the stationary nozzles the design is known as the *Impulse Turbine.* When, on the other hand, there is nearly equal drop of pressure in both the fixed and moving blades as the steam is passing through them the design is generally referred to as a *Reaction Turbine.*

These names, Impulse and Reaction, have been adopted from the analogy of the water turbine, though like a number of other terms current in engineering practice, they cannot be said to be a logical definition of what actually takes place in either type. To a less extent, the same objection holds good to the alternative definitions of velocity type and pressure type which occur in a number of text-books, but which are not commonly used in practice.

The Impulse Type.—In the impulse turbine, by suitably proportioning the cross-sectional areas of the nozzles, the increase in volume or

[1] Baumann, "Recent Developments in Steam Turbine Practice," *Journal Inst. E.E.* vol. xlviii. (1912), p. 827. In a more recent paper on the same subject (*ibid.* vol. lix. (1921), p. 580), Baumann points out that a total steam temperature of 700° F. is now common, and that pressures of 350 lb. sq. in. (gauge) are being adopted in new power stations.

[2] "Correction Tables for Thermodynamic Efficiency," 1917, published by Edward Arnold.

expansion due to the pressure drop causes the steam to issue with a comparatively high velocity. The moving blades are so curved that as much as is practicable of this high velocity is absorbed in revolving the wheel, and the steam leaves with considerably less velocity, but the pressure on both sides of the moving blades remains practically unaltered. Each set of expanding nozzles, together with their wheel, which may have one or more rows of blades, is known as *one stage* of the turbine. If the impulse turbine has only one stage, as in the cases of the single wheel with multiple rows, or of the De Laval type with one row, the pressure drops in the expanding nozzles to that of the exhaust, or nearly so. The moving blade passages are so shaped that the exit velocity of the steam is just sufficient to clear the wheel. If there are two or more stages, such as are found in the Rateau, Zoelly, and Curtis types, the pressure drops in steps through each stage, but remains the same or nearly the same on each side of the moving blades.

The Axial Flow Reaction Type.—In axial flow reaction turbines, such as the Parsons type, the high-pressure steam first enters a row of fixed blades, in which it is caused to expand slightly, and at the same time to increase in velocity. The steam is then passed through a row of moving blades, so proportioned that whilst the increase of velocity is absorbed, the pressure also diminishes as much as it did in the previous row of fixed blades. This combination of one row of fixed blading and one row of moving blades forms *one stage* of a reaction turbine, and there are always a large number of such stages as compared to the impulse type. The pressure falls gradually throughout the length of the turbine, and as the specific volume of the steam is correspondingly increasing the diameter of the drum carrying the blades is stepped up to prevent the blade heights from becoming excessive. Each diameter is referred to as a *drum*, so that a reaction turbine often consists of *a high-pressure drum*, *an intermediate-pressure drum*, *and a low-pressure drum*, on each of which there is an appropriate amount of stages.

In very large turbines the low-pressure drum is sometimes made as a separate machine in which the steam generally enters at the middle and divides right and left to exhaust at each end. In this way the blade heights are halved. Such a machine is known as a *double-flow* turbine. In a few cases, the double-flow low-pressure drum is included in one casing or cylinder with the high-pressure drum.

For mechanical reasons of construction the heights of the blades are kept the same for a number of stages, but they may be stepped up on each drum to approximate more closely to the conical reaction turbine which is the ideal shape for this type not yet completely achieved in practice.

Combination Reaction and Impulse Turbine.—In the impulse turbine the wheels carrying the moving blades are frequently referred to as *discs*, whilst the cylindrical spindle on which the moving blades of the reaction turbine are mounted is called the *drum*, and many modern turbines which form a combination of the two are therefore known as the *disc and drum type.* In such a case, a number of stages of reaction blading at the high-pressure end of the turbine are replaced by one or more

impulse stages, with a considerable saving in the overall length of the turbine.

The Radial Flow Reaction Turbine.—Nearly all successful turbines of the present-day work with the steam flowing axially, that is to say parallel with the shaft or spindle. There is, however, one notable exception, the Ljungström turbine, in which the steam enters the first stage near the spindle and flows radially outwards at right angles to the main axis of the turbine. As expansion takes place in moving blades this turbine belongs to the reaction type. In this turbine there are two discs, each carrying reaction blades, which project from them axially, and which are connected to them by specially shaped expansion rings. The two discs revolve in opposite ways, so that the relative speed of the rings of blades to one another is doubled, which enables the whole of the expansion to be carried out in a single pair of discs. This turbine in large sizes may have one or more rows of Parsons blading at the exhaust end, that is to say, an axial flow exhaust.

Turbines for Special Purposes.—The requirements of a number of industries for steam other than for power generation has resulted in the design of modified types of steam turbines, both impulse and reaction, which have gradually come to be known by distinguishing names. For instance, a turbine may be made to work non-condensing to exhaust either at atmospheric pressure or above it. This enables a constant supply of low-pressure steam to be available for manufacturing purposes. Such a turbine is often called a *back-pressure turbine*. More commonly the demand for low-pressure steam is not regular, but varies from time to time, as in heating the factory, which is a winter demand, or in a number of manufacturing processes which are intermittent. In that case the turbine is so arranged that steam may be drawn off as required from a point just above atmospheric pressure, whilst the remainder of the steam passes through the low-pressure end of the turbine to the condenser in the usual way. The name *Reducing Turbine* is applied to such a design, though in the United States of America they are more appropriately called *Bleeder* or *Extraction Turbines*.[1]

In a reciprocating engine on account of fluid friction, and the size of low-pressure cylinder which would be required, it is rarely possible to carry the expansion down to the vacuum in the condenser, and release generally takes place at 2 or 3 lb. per sq. in. above the pressure in the condenser. Thus the whole of the energy in the last part of the expansion is lost in such an engine, and the extra heat drop due to high vacua is not available. As a result vacua for reciprocating engines are generally between 25 and 27 in. and are rarely higher. In the case of a steam turbine, however, by suitably proportioning the exhaust blades, the very highest vacuum possible in practice can, as a rule, be made use of, and, therefore, vacua of from 28½ to 29 in. or even over are common. The great advantage that a steam turbine has in being able to utilise the highest vacuum possible is at once seen if the heat drop available by Peabody's tables, from, say, saturated steam at 215 lb. absolute to various vacua, is considered.

[1] For an account of modern reducing turbines, see paper by H. L. Guy, "Steam Turbines for Land Purposes," *Trans. Manchester Association of Engineers*, 1916–17, p. 97.

To 25 in.	vacuum	294	B.Th.U.	available
26 in.	,,	306	,,	,,
27 in.	,,	320	,,	,,
28 in.	,,	343	,,	,,
29 in.	,,	370	,,	,,

Here it is seen that between, say, 27 in. and 29 in. there is a difference of 50 B.Th.U., or, say, 15 per cent.

This capacity of the steam turbine for utilising high vacua has led to the introduction of *Exhaust Turbines* where the steam from a reciprocating engine, exhausting at about atmospheric pressure, is further utilised, and, as a rule, the power derived from the turbine is about the same as from the reciprocating engine, and thus the power obtained with a given amount of steam is doubled. In many cases the reciprocating engine and the steam turbine form one unit, but if the reciprocating engine works intermittently, as in the case of a rolling mill engine or a winding engine, a thermal accumulator is fitted between the two to give an approximately constant supply of steam to the steam turbine. In some cases, however, the supply of exhaust steam is at certain times liable to fail or not be sufficient to supply the turbine, and then a high-pressure stage is often fitted to which high-pressure steam is automatically turned on when the low-pressure supply fails, or is not sufficient. The turbine is then called a *Mixed-pressure Turbine.*

Definitions—Reciprocating Steam Engines.—The history of the steam engine is so much older, and the accumulated knowledge is so much more widely diffused, than that of other heat engines, that it is only necessary here to touch on a few of the salient points in connection with steam engines in modern practice. The principle of expanding steam behind a moving piston was the only practical means of obtaining power from heat engines that was used by three generations of engineers, and the impetus they gave to this method has carried it well over the forty years that have now lapsed since the internal combustion engine and the steam turbine appeared on the horizon.

In its essentials, steam is led from a boiler in steam pipes to a *stop valve*, which is usually situated close to the engine, and passes into and out of the *cylinder* containing the *piston* through valves which are placed in a *valve chest* on the cylinder itself. The initial pressure and temperature of the steam should be taken on the boiler side of the stop valve when the thermal efficiency is to be obtained. The valves in the valve chest are actuated by the engine itself, and can be timed to regulate the *cut-off* of the steam and the period of exhaust. With very few exceptions steam is admitted in this way to both sides of the piston in turn, so that the steam engine is *double-acting* and receives *two impulses per revolution.* This requires that the piston rod should pass out of the cylinder through a steam-tight *gland* or *stuffing box.* The outside end of the piston rod is fitted to a *crosshead* which works in some form of *slide*, and the reciprocating motion is converted into a rotary motion through a *connecting rod*, the *small end* of which works on the crosshead pin, and the *big end* of which drives the *crank* through the *crank pin.* The other end of the crank is fixed to, or forms part of, the *crank shaft* from which the power of the engine is obtained.

The *flywheel* when used is mounted on the crank shaft, and serves the double purpose of flattening out the turning moment on the crank pin, and keeping the variation in speed per revolution of the crank pin within prescribed limits. A *governor* is required which controls the complete revolutions made by the engine per minute. Both the flywheel and the governor may be absent in marine and locomotive steam engines. The position and number of cylinders in one unit have given rise to various types of engines. A *simple engine* has only one cylinder, but if the steam is expanded in two or more cylinders in succession it is known as a *compound*, *triple expansion*, or *quadruple expansion* engine. Not infrequently the specific volume of the steam is so large by the time that the last stage is reached that two low-pressure cylinders are used side by side in triple expansion engines. The position of the cylinders has given rise to the terms *horizontal* and *vertical* engines. If two cylinders are placed one behind the other on one piston rod the engine is called a *tandem*, whilst if the high-pressure and low-pressure cylinders are side by side with the flywheel in between the type is known as the *cross-compound.*

Cylinders are sometimes surrounded by steam to keep them warm. They are then said to be *jacketed.* Innumerable designs of valves and valve gear have been successfully used, but the main types in use at the present day are the *slide valve* and the *piston valve*, which are usually driven direct by *eccentrics* on the crank shaft, or the *Corliss valve* and the *drop valve*, which are closed by springs released by a trip action.

The Still Engine.—For some years now W. J. Still has been working on attempts to combine the superior thermal efficiency of the internal combustion engine cycle with the flexibility and mechanical advantages of the steam engine.[1]

The idea underlying the Still engine is to have, in one cylinder, steam on one side of the piston and some form of internal combustion on the other.

Whilst the range of temperature through which internal combustion engines work is much greater than is possible with steam, the lowest temperature or heat rejected to exhaust is still comparatively high, and many attempts have been made to utilise part of this heat. The lines upon which Still and his associates have been working include not only a feed-water heater in the exhaust of an internal combustion engine, but also an arrangement of the cooling jacket surrounding the engine cylinder in such a way that the cooling effect is not produced by raising the sensible heat and therefore the temperature of the water, but by utilising the latent heat of water during its conversion into steam *without* affecting its temperature.

This is obtained by making the jacket and the cooling water part of the circulating system of a steam generator, so that the temperature is controlled by the pressure of the steam in that system, and the cooling is obtained by the conversion of water into steam without raising its temperature. An example of this type of engine is shown in the internal combustion engine section.

[1] See paper read by F. D. Acland before the Royal Society of Arts, on May 26, 1919. There is an illustrated abstract of this paper in *The Engineer*, vol. cxxvii. (1919), p. 540.

CHAPTER VI

DESCRIPTIONS OF STEAM TURBINES AND STEAM ENGINES

Steam Turbine Types.—Various classifications of steam turbines have been put forward by different authorities, but in practice they all fall into two groups, *impulse* and *reaction*, or into a combination of the two, usually known as the *disc and drum*.

In this chapter the following types are selected as representative examples of modern practice :—

Impulse Turbines

Single-wheel turbine with—

(*a*) 1 row of blades. De Laval.
(*b*) 2 rows of blades. Brown Boveri.
(*c*) 3 rows of blades. Escher Wyss.

Multiple-wheel turbine with—

(*a*) single row of blades on each wheel. Zoelly.
(*b*) 2 rows of blades on first wheel, followed by single-row wheels. Metropolitan-Vickers, Fraser & Chalmers.

Reaction Turbines

(*a*) axial-flow reaction turbine. Parsons.
(*b*) radial-flow reaction turbine. Brush-Ljungström.

Disc and Drum Turbines

(*a*) 2-row impulse wheel, followed by two sections of reaction blading. Richardsons, Westgarth.
(*b*) three 1-row impulse wheels, followed by reaction blading. Brown Boveri.

Impulse Turbines.—*The De Laval Steam Turbine* has for a long time been made in this country by the firm of Greenwood & Batley, of Leeds. The characteristic of the design lies in the use of a single wheel with one row of moving blades and reduction gearing to bring the speed down to useable values. The turbine is made by this firm in 16 standard sizes, varying from 3 to 600 B.H.P. ($1\frac{5}{8}$ – 400 K.W.) and a section

through a 225 B.H.P. machine is shown in Fig. 54.[1] Steam is admitted through a stop-valve, strainer, and governor valve to groups of nozzles drilled in a ring, which is fixed to the turbine housing. These groups, each containing two or three nozzles, are arranged symmetrically round the turbine wheel. Each group has a separate shutting-off valve to provide hand-adjustment for partial loads. In the smaller sizes, each

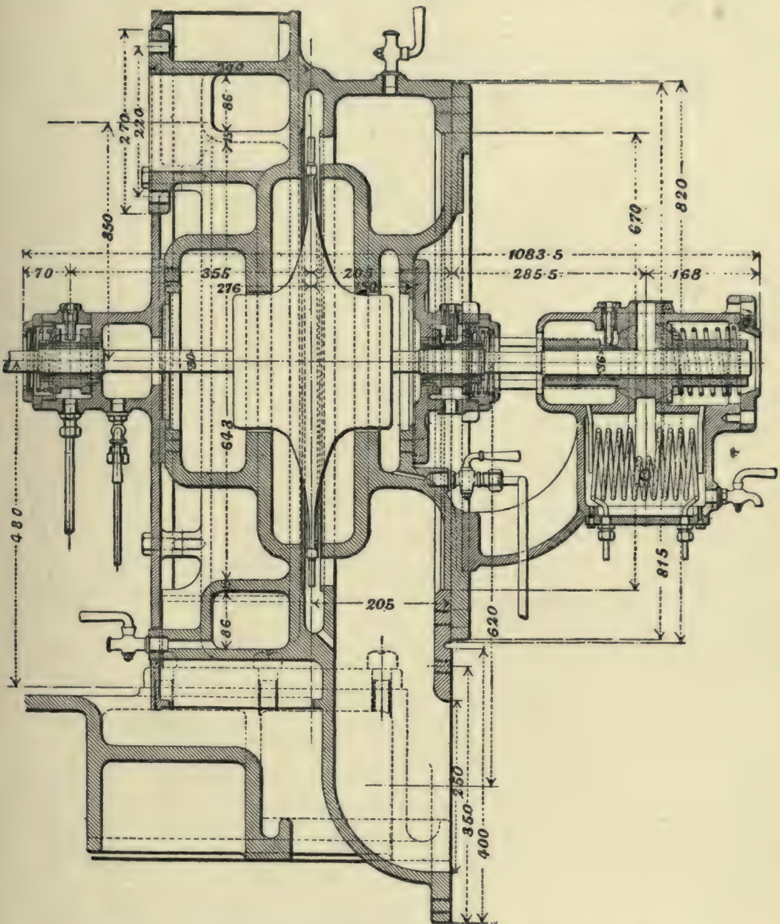


FIG. 54.—Small impulse turbine with 1-row wheel (De Laval).

nozzle has a separate control valve for the same purpose. The steam is expanded in the nozzles practically to the back pressure on the turbine, and the wheel is driven by the velocity of the steam due to this total heat drop. The speed of the wheel in the smallest size is 30,000 R.P.M., and in the largest about 10,000 R.P.M.; but the

[1] Reproduced from Neilson's "Steam Turbines."

gearing reduces these values to 3000 and 750 R.P.M. respectively. Such high speeds are well above the "critical" speed of the rotor, and the shaft is kept as small in diameter as possible to make it flexible. The bearings, one only of which is seen outside the right-hand stuffing gland, are purposely set some distance apart for the same reason. The rotor is thus able to take up its own balanced position after the critical speed is passed and run perfectly smoothly. The shaft is connected to a gear-box, which contains two pinions supported on three bearings. The lubricating oil is pumped out of a chamber underneath the gearbox, and this oil reservoir is cooled by a number of pipe coils through which cold water is circulated. The main governing is by throttle actuated through a rocking lever and a centrifugal governor mounted on the shaft of the gear wheel. In the case of condensing turbines, should the governor valve fail, or allow the speed to increase more than 5 per cent., an auxiliary governor air-valve is used, which reduces the vacuum automatically and drops the speed. If the turbine is exhausting into a condenser with a large volume, or into a common condenser in which it would not be convenient to reduce the vacuum, the air-valve operates on a vacuum governor, instead of direct into the condenser, which diminishes the vacuum in the exhaust chest of the turbine instead. The high peripheral disc speeds necessitate a special design of rotor wheel above the 20 B.H.P. size. This design can be seen in Fig. 55, which is kindly supplied by the makers. The wheel is made solid, without any central hole, and has a concavo-convex profile. The shape is calculated to give equal stresses from the centre to the rim. As a safety device the thickness just under the rim is then reduced to ensure fracture at that portion of the wheel in case of accident. The shaft is divided into two and fixed to each side of the disc by flanges and studs. The whole arrangement greatly strengthens the disc as compared with one having a central hole, and enables peripheral speeds of 1300 ft. per sec. and over to be used with safety. The blade and its method of attachment can also be seen in this figure. Any blade can be inserted or removed without disturbing the others.[1]

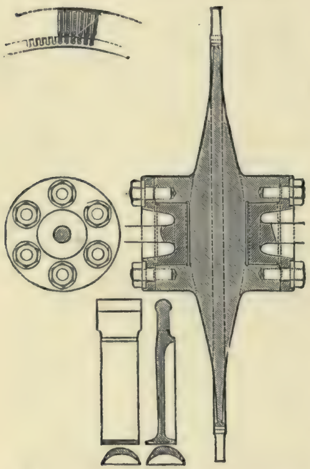


FIG. 55.—Solid De Laval wheel and method of blade fastening.

A modern example of a single-wheel impulse turbine with two rows of blades is shown in Fig. 56. The design is made for powers between 25 and 100 K.W., and speeds varying from 6000 to 3000 R.P.M. It is kindly supplied by the firm of Brown Boveri, of Baden, in Switzerland. It will be noted that the housing is shaped round the disc to reduce windage loss as much as possible. The bearings are

[1] For tests and more details of the De Laval turbine, see H. M. Martin, "Design and Construction of Steam Turbines," 1913, p. 221.

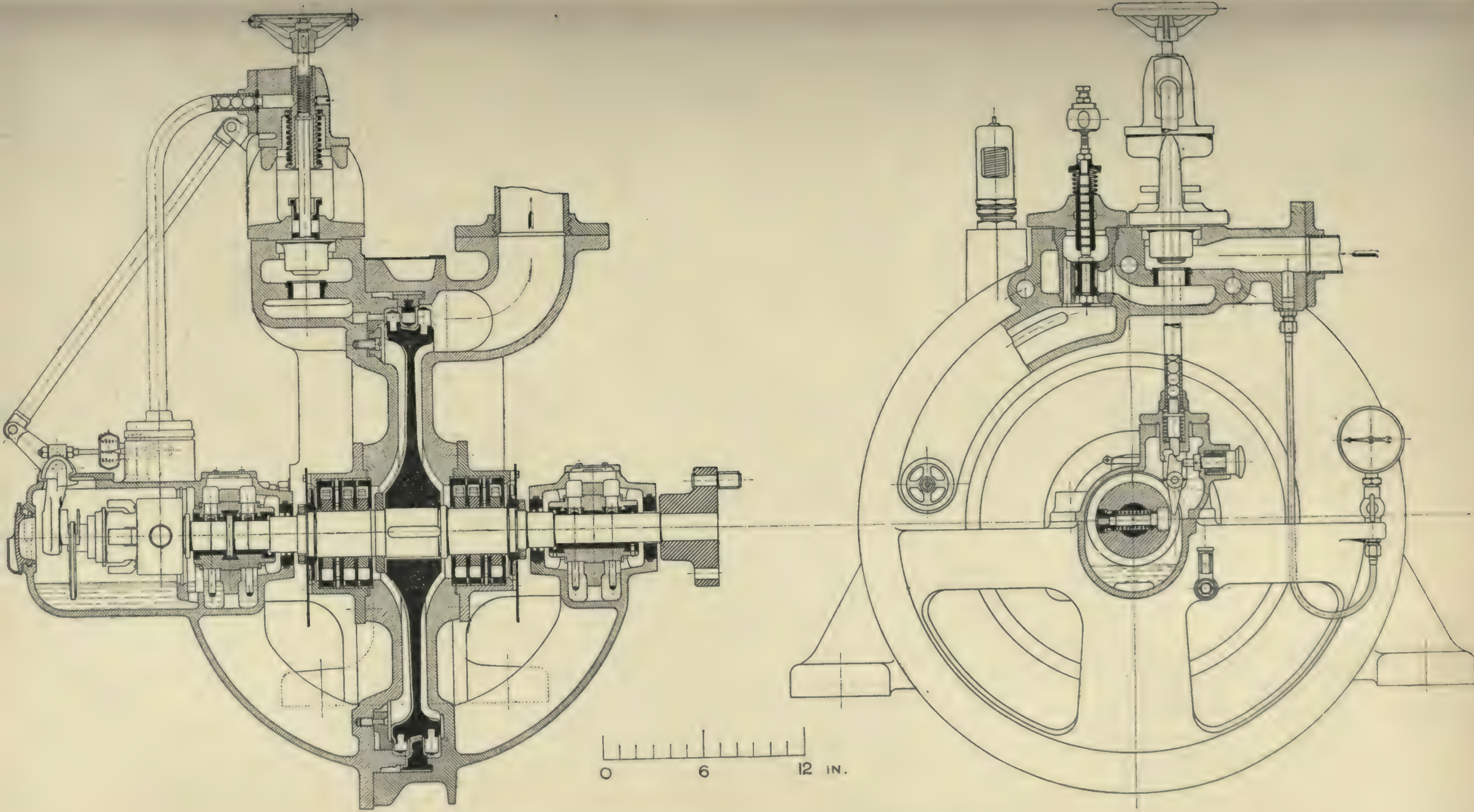


FIG. 56.—Small impulse turbine with 2-row wheel (Brown Boveri).

ring lubricated, and ample carbon glands are interposed to cope with back pressures up to 3 atmospheres should the turbine run non-condensing. The governor on the end of the shaft is directly connected through a rod to the admission valve of the nozzle box. Before reaching this valve the steam has to pass the main stop valve, which can be cut off by a trip action through a number of steel balls in a hollow tube. A regulator, seen above the main governor, can be adjusted to set this trip at the desired overspeed.

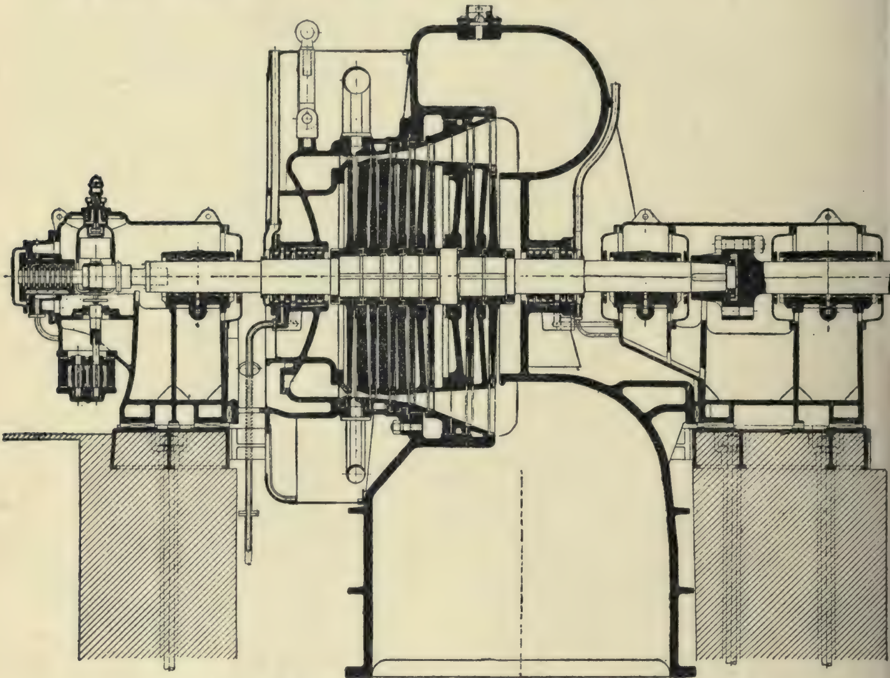


FIG. 57.—Large modern impulse turbine with single-row wheels (Zoelly).

Single-wheel impulse turbines are also made with three rows of moving blades.[1]

Large Impulse Turbines.—A longitudinal section through a modern type of impulse turbine is shown in Fig. 57.[2] It represents a Zoelly turbine, as made by the firm of Escher Wyss & Co., at Zurich, and is included here by their courtesy. The size illustrated is designed to develop 15,000 H.P. when running at 3000 R.P.M. There are seven stages, each consisting of a row of nozzles, followed by a single row of moving blades. The nozzles are formed by a number of nickel steel blades cast into split diaphragms, or, in the case of the first stage, in a ring bolted to the steam chest. The cross-sectional area of the steam passage is rectilinear. In the earlier stages the nozzle arc would

[1] See Goudie, "Steam Turbines," 1917, p. 20, for an example by the firm of Escher Wyss, Zurich.

[2] Reproduced from "The Dictionary of Applied Physics" (Macmillan & Co.).

not extend completely round the circumference, but would be arranged symmetrically in segments. This is known as partial admission. After full peripheral admission is reached, the blade heights are increased to correspond with the increasing volume of the steam. The revolving blades are also made of nickel steel, and fixed to the periphery of steel discs or wheels. These wheels are mounted on expanding rings which are keyed on to the shaft, and kept in position by nuts at each end. These nuts and sleeves tighten up against the collar, which can be seen on the shaft. For turbines running at 1500 R.P.M. or under this shaft is made rigid, and run considerably below the first critical or whirling speed, but at 3000 R.P.M. a flexible shaft is used which runs through the critical under no load during the process of warming up the turbine. Experience shows that this procedure is possible, providing the working speed is kept at not less than 30 per cent. above or below any critical speed of the rotor. The flexible shaft has not entailed the use of a spherical seating for the shaft bearings. These are two in number and are lined with white metal. They are lubricated by oil under pressure. As there is little or no change in pressure whilst the steam is passing through the moving blades, there is, theoretically, no axial thrust along the shaft of an impulse turbine. It is, however, customary to fit a thrust block, shown on an extension of the shaft at the high-pressure end, which is usually made adjustable, so as to register and keep the correct position of the rotor relative to the casing. Such blocks can then take up any slight axial thrust in either direction. It will be noted that the casing as a whole is not rigidly fixed to the two main bearings, but that these latter are mounted in separate housings on two cross girders embedded in the foundation. The casing is carried on sliding supports which allow for any expansion due to heating effects. The bearing housing can also move axially, so that provision is made for the turbine to adjust itself relative to the centre line of the exhaust, which is fixed by the condenser.

The main inlet of the steam is not shown. It is admitted through a throttle governor operated by an oil relay into one side of the annular space to which the first row of nozzles is bolted. The steam passes through strainers to the nozzle segments arranged symmetrically in front of the first row of moving blades. A supplementary steam inlet is provided for overloads, in this case between the first and second stages. The steam then passes through the remaining stages into the exhaust branch which surrounds the last row of blades, and from there to the condenser underneath the turbine. The shaft, as it enters and leaves the turbine casing, passes through carbon-packed glands, which effectively prevent any steam from escaping on the live side and maintain the required vacuum at the exhaust end. In order to prevent leakage from stage to stage at the shaft, it is necessary to keep down the clearance between the fixed diaphragms and the revolving wheel-hubs as much as possible. At the same time a certain amount of latitude must be allowed for any slight deflection of the shaft. If the pressure drop is considerable it is advisable to use some form of flexible gland, such as carbon segments held in position by springs, but in most cases the difficulty is surmounted by leaving a space between the diaphragm and the hub, and inserting a number of wedge-

shaped soft metal rings on the inner circumference of the diaphragm, with their apex towards the shaft, so that, should touching occur, no material damage will ensue.

Another example of a modern impulse turbine is shown in Fig. 58.[1] It is reproduced by kind permission of the Metropolitan-Vickers Electrical Co., of Manchester, and shows a 2-row wheel followed by 13 single-row wheels. The normal output is 12,500 Kilowatts, and the speed 3000 R.P.M. Instead of allowing the steam to fill the annular space in front of the first stage, it is admitted into one, two, or three separate nozzle boxes, for half load, full load, or overload respectively. These boxes, which are made of cast steel, are so shaped as to be free to expand without affecting the alignment of the nozzles which form part of the first stage. As in the Zoelly turbine, the main bearings are housed separately from the casing, and provision is made to allow for any expansion or distortion of the materials of the turbine under the heating effects of the steam. An arrangement of multi-exhaust blading is shown, which is sometimes embodied in impulse turbines made by this company. It is designed to obviate the difficulty of dealing with large quantities of steam at very low pressures, which would otherwise entail a larger diameter of blade ring, or the use of excessive lengths for the exhaust blades. The moving blade in the last stage but two is divided into two portions. The outer half is shaped to allow the steam passing through it to expand to the pressure of the exhaust. The remainder of the steam is bye-passed through the inner half without expansion, and again divided up by means of a specially-shaped fixed nozzle to repeat the process through the next row of moving blades. In the last stage all the remaining steam expands to the last row of moving blades, and passes through to the exhaust. The effect produced may be taken as the equivalent of a blade height equal to the sum of the two outer portions of the divided blades, plus the length of the moving blade in the final stage.

Another method of compounding the exhaust is shown in Fig. 59. This turbine, the details of which are kindly supplied by Messrs. Fraser & Chalmers, of Erith, in Kent, is also of the impulse type with a single 2-row wheel, followed by 15 1-row wheels. It is designed to give 20,000 K.W. when running at 1500 R.P.M., and is suitable for a steam pressure of 300 lb. per sq. in., a total steam temperature of 700° F., and a vacuum of 29·1 in. with a 30-in. barometer. The low-pressure end is designed for double flow. About one-third of the steam passes into the condenser through the three stages immediately following the single-flow portions, whilst the rest is bye-passed to the two larger wheels at the end of the rotor.

The three smaller wheels and the two larger wheels work in parallel. The smaller diameter of the former making the efficiency of the blading in both portions about the same.[2]

Reaction Turbines.—Fig. 60[1] shows a modern example of a 10,000 K.W. high-pressure tandem reaction turbine kindly supplied by the firm of C. A. Parsons & Co., at Newcastle-upon-Tyne. It is

[1] Reproduced from "The Dictionary of Applied Physics" (Macmillan & Co.).

[2] For a detailed description with drawings of a 2000 K.W. mixed-pressure turbine by the same firm, see *Engineering*, vol. c. (1915), p. 49.

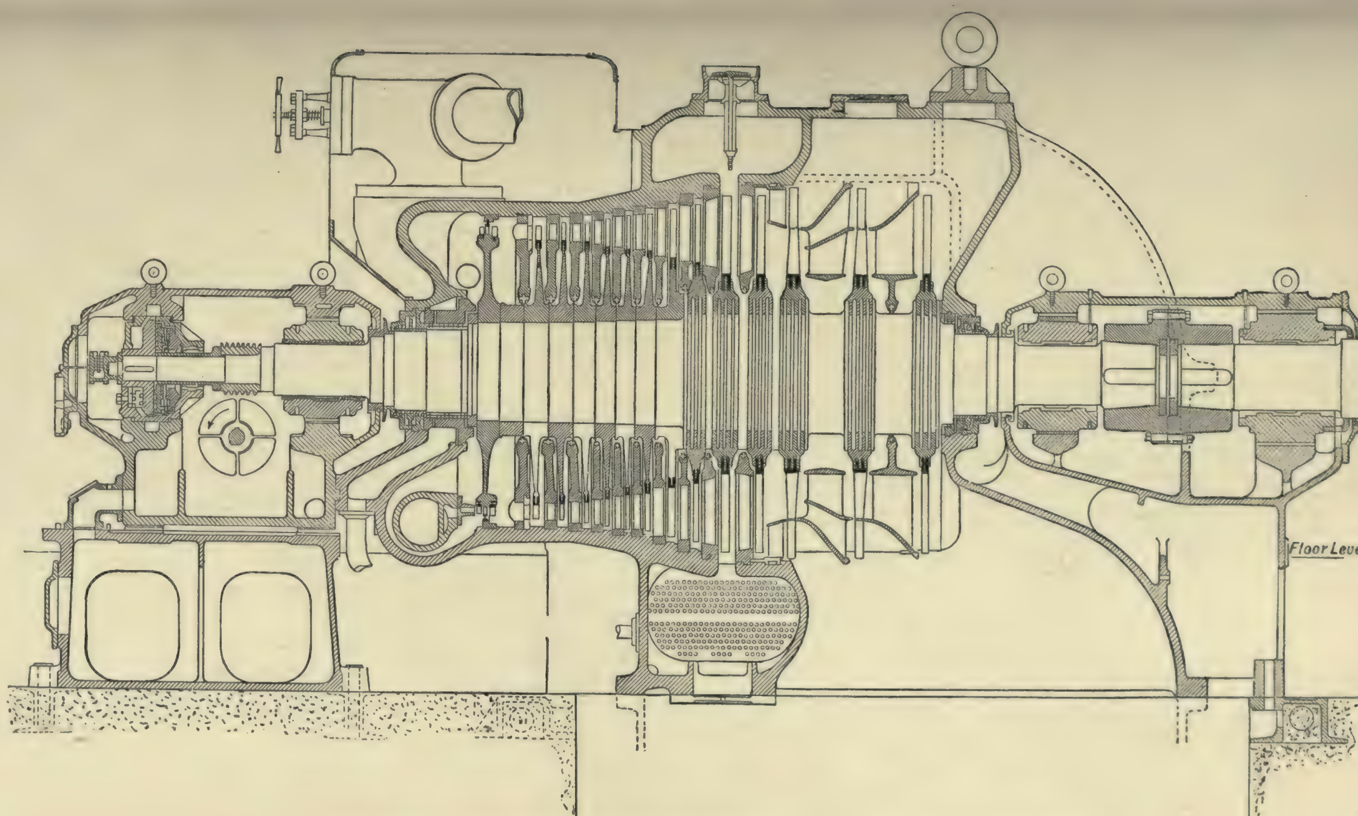


FIG. 58.—Modern large impulse turbine with multiple exhaust (Metropolitan-Vickers).

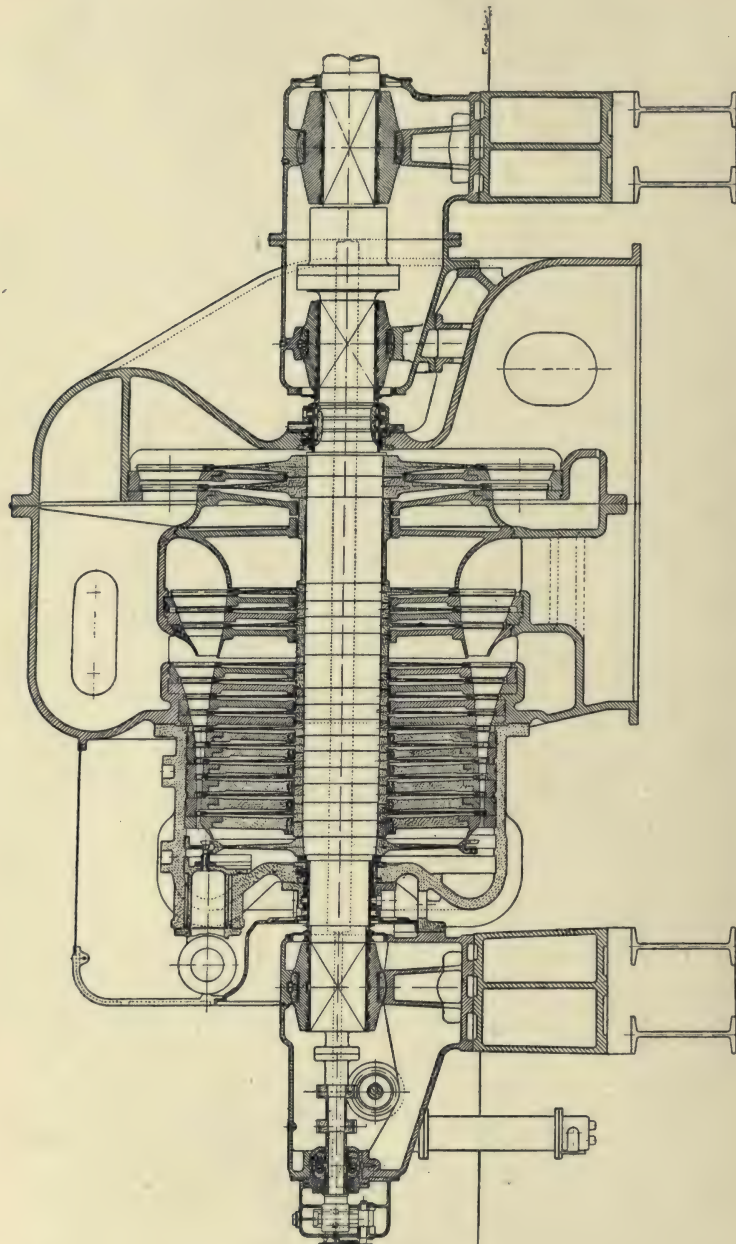


Fig. 59.—Modern large impulse turbine with multiple exhaust (Fraser & Chalmers).

designed to run at 3000 R.P.M. The steam, after passing through the main inlet valve (not shown), enters the annular space surrounding the first stage from underneath. Overloads are automatically taken up by the action of the bye-pass valve shown above this space, which admits the full-pressure steam to a second annular space surrounding the second section of stages. The steam then passes through the remaining rows of blading in the first turbine, and is carried by a steam pipe connected to the large flange shown below the second main bearing to the centre of the second turbine, where it divides to pass through more stages and exhaust at both ends into a condenser connected to the two exhaust branches. The low-pressure drum is therefore balanced, but the first turbine requires *balance pistons* or *dummies* to counteract the axial thrust on the moving blades. These dummies are shown to the left of the first stage. They are three in number, one for each diameter of the turbine drum, and connecting pipes (not shown) maintain an equal pressure between the dummy and its corresponding section of the turbine. It will be noted that each turbine has an end-thrust bearing fitted in front of the left-hand main bearing in each case. This ensures

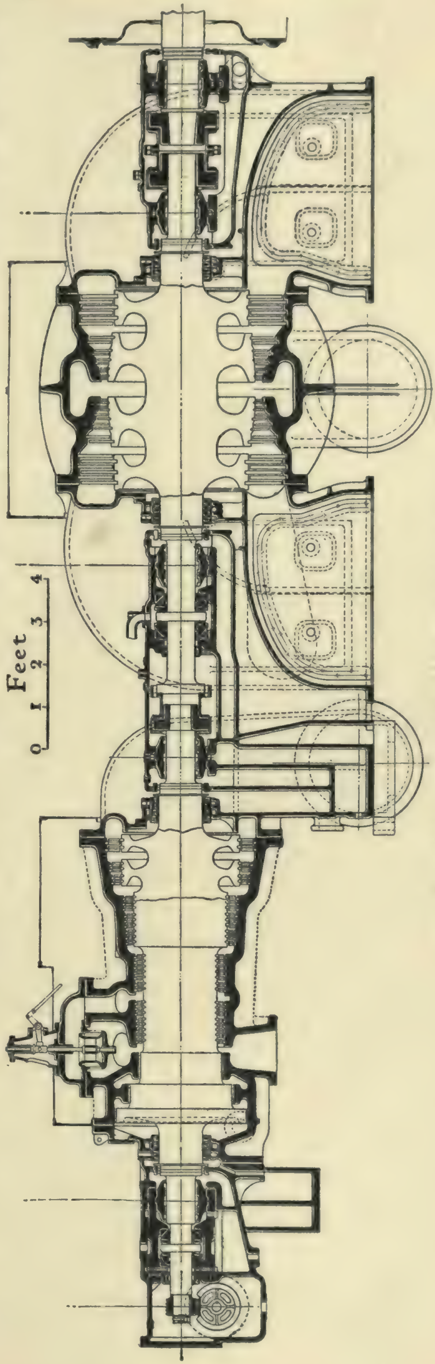


FIG. 60.—Modern axial-flow reaction turbine (Parsons).

a correct register of the fixed and moving blades as in the impulse type.

An example of a 1500 K.W. Ljungström turbine is shown in Fig. 61. It is made in England by the Brush Electrical Engineering Co., of Loughborough, who have kindly supplied Figs. 61 and 62.[1] High-pressure steam enters through a pipe which is brought into the exhaust and passed by two expansion joints to annular steam chests, surrounding the labyrinth packing gland on the end of each shaft. From these annular spaces the steam has direct access to the centre of the turbine where it enters the first stage and flows radially to the periphery of turbine. Arrangements are made for bye-passing live steam into the turbine after the first few stages, to cope with overloads. In order to curtail excessive blade lengths the last few stages are arranged for parallel flow. In the larger machines the last stage may be carried out in axial flow reaction blading of the Parsons type. Such an arrangement can be seen in Fig. 62, which is a compounded section of half the turbine as far as the shaft. The right half of this drawing shows the upper part of the turbine, and details of the bye-pass for overloads. The left half shows the underneath part with the main steam inlet. Elaborate precautions are taken to allow for the expansion of all parts subject to high temperature, without affecting their relative positions. This is satisfactorily achieved by the use of a number of expansion rings, whose cross section resembles somewhat that of a dumb-bell. The metal in the disc or blade ring, as the case may be, is closed round the head of this dumb-bell by rolling to form a circular socket joint which is firm but flexible. It will be noted that all parts exposed to high-temperature steam, including the radial labyrinth packing and the steam chest itself, are linked up to the outer casing by means of these expansion rings. The radial labyrinth glands are so proportioned that they balance any axial thrust of the steam tending to force the blade discs apart.

Each shaft is direct coupled to an alternator, and the stator windings of the two alternators are permanently connected in parallel, so that electrically the two machines form a single unit, though mechanically they are revolving in opposite directions at half the speed of the relative velocity of the blades to one another.

The Ljungström turbine belongs to the reaction type since there is a drop in pressure in each of the two blades forming one stage.[2]

Disc and Drum Types.—A Disc and Drum turbine, as made by the firm of Richardsons, Westgarth & Co., of Hartlepool, is illustrated by their kind permission in Fig. 63.[3] It has an output of 20,000 K.W. when running at 1500 R.P.M. The turbine consists of a 2-row impulse wheel followed by two sections of reaction blading. These reaction stages approximate closely to the conical type of turbine.[4] They are balanced by one dummy piston, seen just to the left of the impulse wheel, whose diameter is approximately the mean

[1] Reproduced from "The Dictionary of Applied Physics" (Macmillan & Co.).

[2] For design of Ljungström turbines, see Goudie, "Steam Turbines," 1917 ed. chap. xv.; see also *Engineering*, vol. xciii. (1912), p. 583.

[3] Reproduced from *The Electrician*.

[4] For a theory of the conical turbine, see the article on "The Physics of the Steam Turbine" in "The Dictionary of Applied Physics."

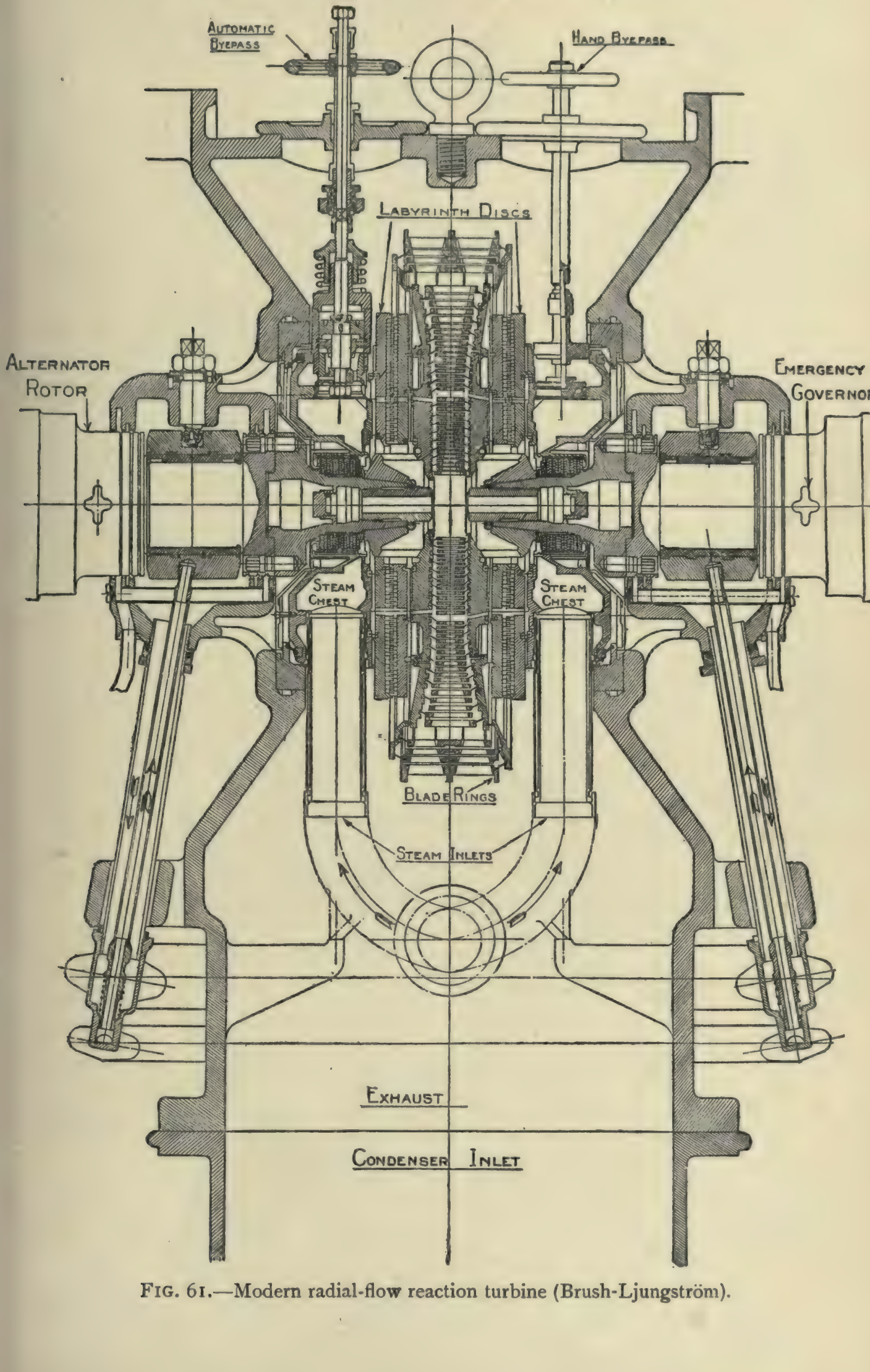


FIG. 61.—Modern radial-flow reaction turbine (Brush-Ljungström).

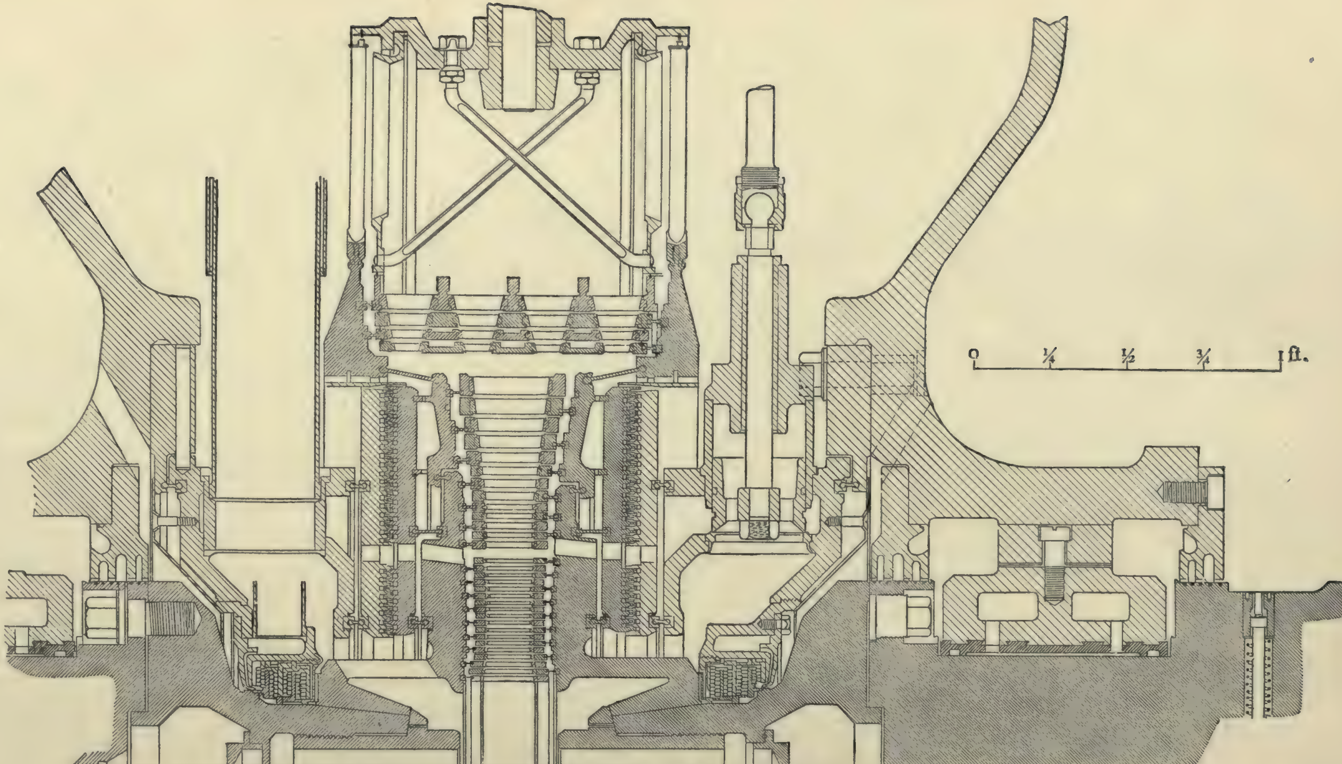


FIG. 62.—Half section of Ljungström turbine with exhaust "Parsons" blading.

diameter of the two reaction sections. Other good features referred to above under the impulse and reaction types are embodied in this design.

A recent modification of the combination of impulse and reaction types is shown in Fig. 64, which has been kindly supplied by the firm of Brown, Boveri & Co., of Baden, in Switzerland. The output of this machine is 15,000 K.W. at 3000 R.P.M., but the design can be used for sizes up to 35,000 K.W. at 1250 R.P.M.

In this design both the high-pressure and the intermediate-pressure drums are replaced by impulse wheels. The first section by a single 1-row wheel and the section following by two further 1-row wheels with intermediate nozzle diaphragms. The low-pressure or reaction part has only four stages in which the fixed blades are made wider than the moving blades, but so shaped that there is approximately an equal drop in pressure through both sections of the stage. This extra width enables discs with stronger hubs to be made than would otherwise be the case, so that the mean peripheral speed can be increased. In this case it is 770 ft. per sec. Since the total pressure drop through the reaction section is probably not more than 15 lb. per sq. in., the drop in pressure through each fixed blade can be maintained by end tightening or running projecting lips round the shrouding of these blades. This does away with the necessity of diaphragms or partitions for the fixed blades. They are simply mounted round the turbine casing, and project inwards.

It will be noted that this arrangement enables the mean diameter of the blades to be kept constant throughout the turbine. The turbine is governed by five nozzle valves operated by oil under pressure which is directly controlled by the governor. The valves open and close one after the other according to the load.

An annular chamber between the impulse and reaction stages has an opening underneath through which steam can be passed. This enables either additional steam to be admitted into the low-pressure end of the turbine or conversely live steam to be extracted for heating or other factory purposes.

The axial thrust of the reaction section is balanced partly by the dummy to be seen on the left of the first impulse wheel and partly by the end thrust bearing beyond the left-hand main bearing.

Reciprocating Steam Engine Types

The vast amount of information available about the design of reciprocating steam engines, renders it unnecessary to include here more than one or two examples as typifying modern progress.

Horizontal Tandem Drop-Valve Engine.—Fig. 65 shows a pair of horizontal drop-valve mill engines, arranged side by side on one rope-drum flywheel with the high-pressure cylinders behind the low-pressure cylinders. This engine, which was described in the *Textile Mercury*,[1] was built by the firm of Petrie & Co., Ltd., of Rochdale, and develops 1400 I.H.P. with 160 lb. per sq. in. (gauge) steam pressure when running at 85 R.P.M. The high-pressure cylinders are 19 in.

[1] *Textile Mercury*, vol. xli. (1909), p. 309.

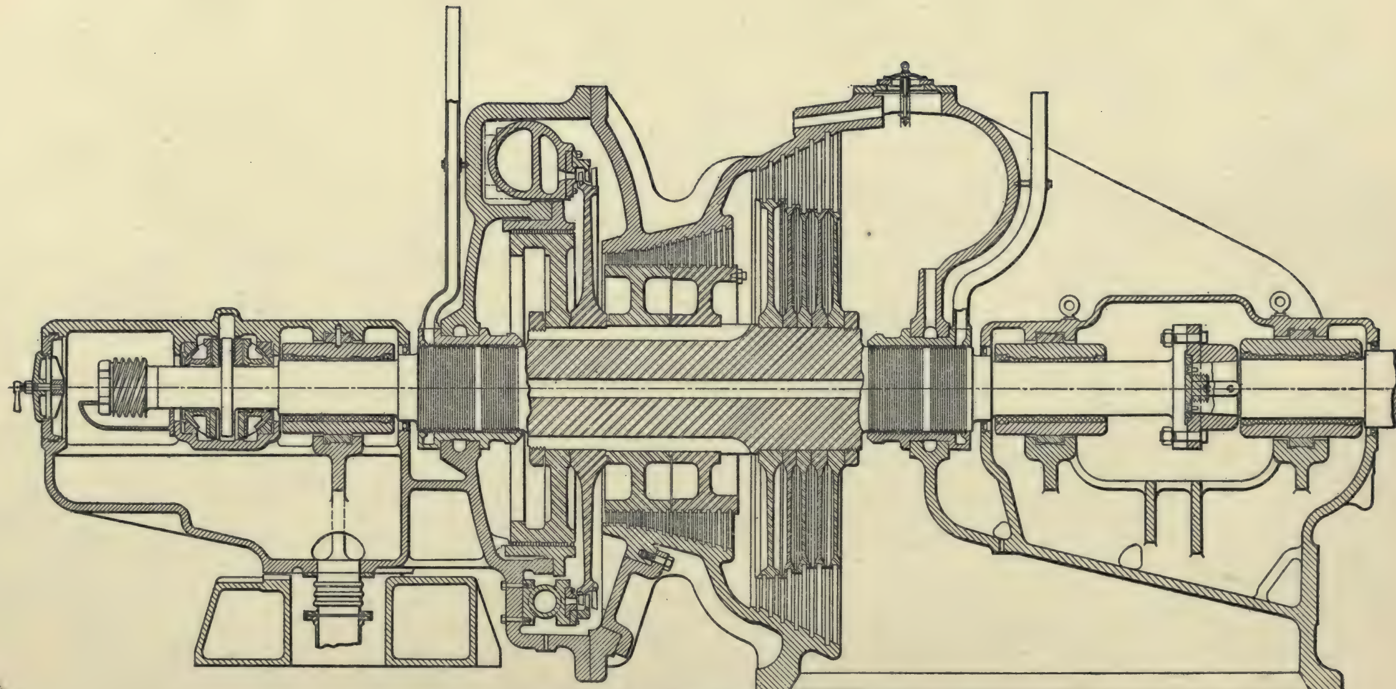


FIG. 63.—Modern disc and drum turbine (Richardsons, Westgarth).

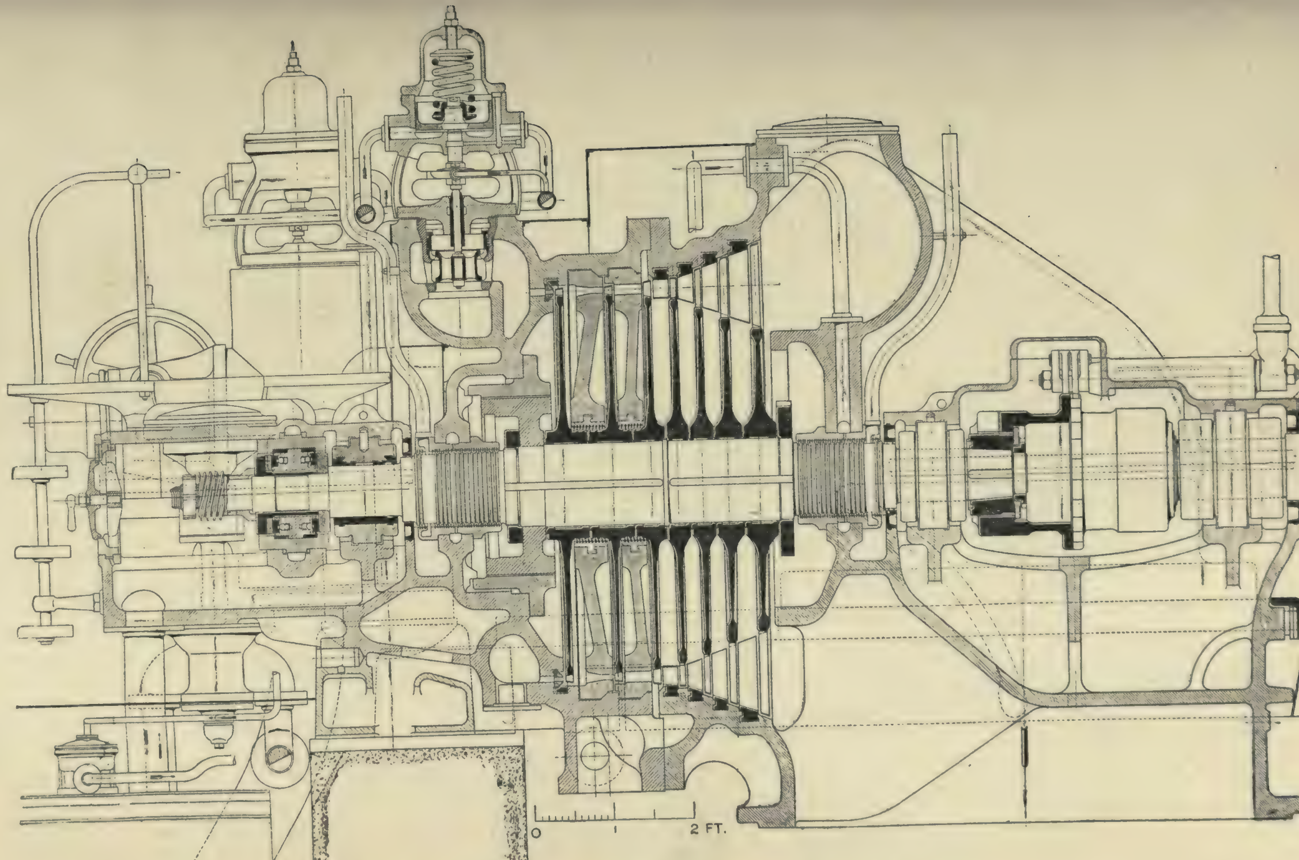
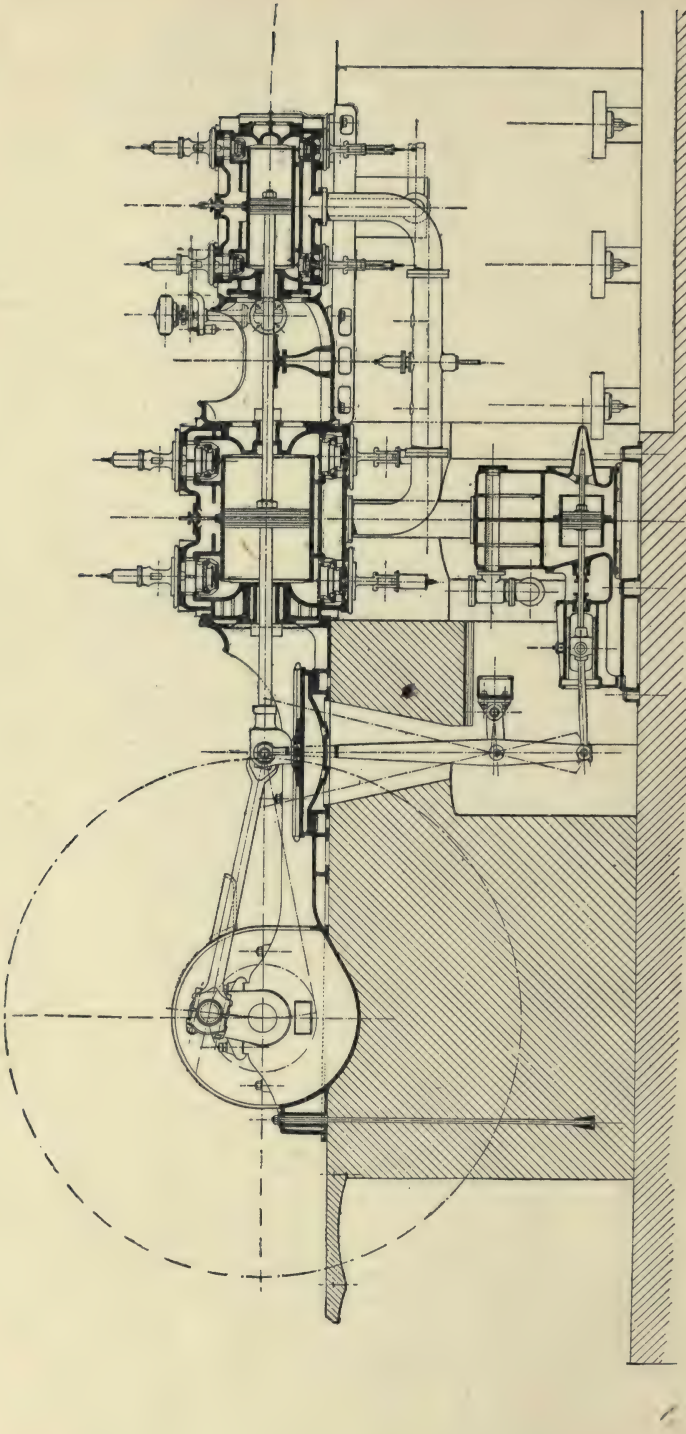


FIG. 64.—Modern large impulse reaction turbine (Brown Boveri).



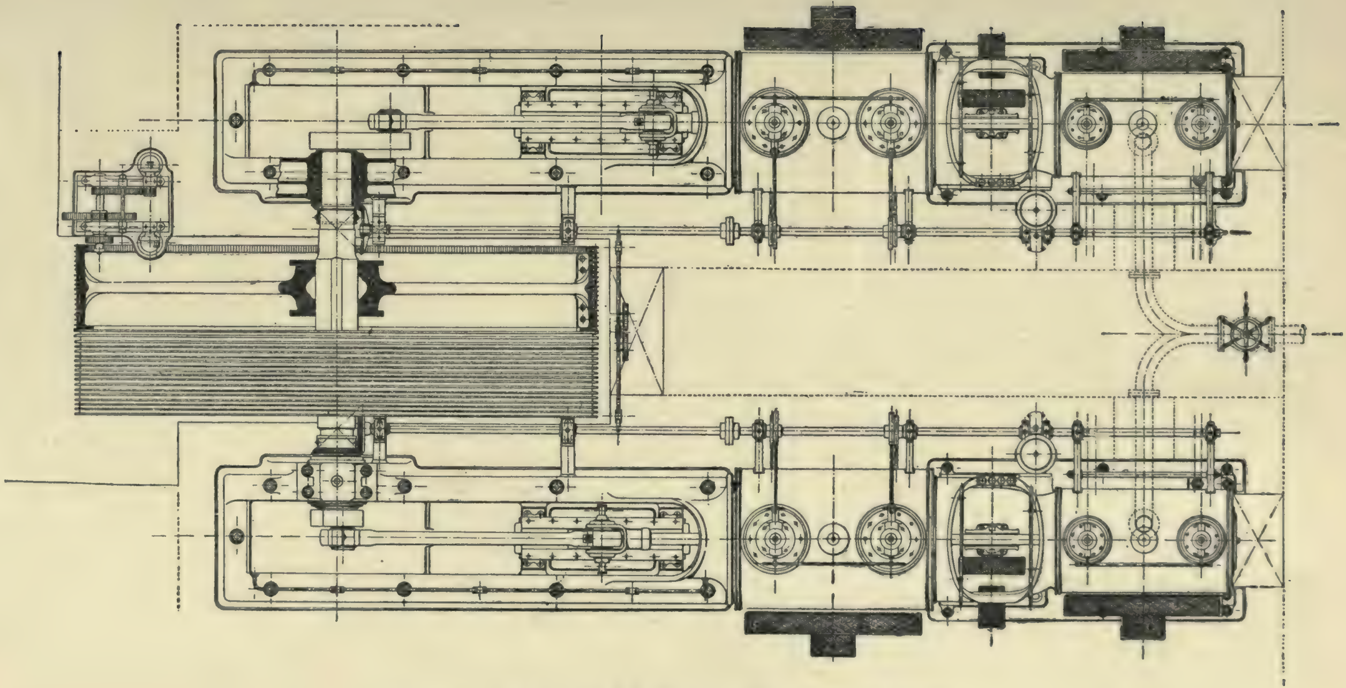


FIG. 65.—Pair of horizontal tandem drop-valve steam engines.

diameter and the low-pressure cylinders 38 in. diameter, the stroke in each case being 4 ft. The steam enters from a common stop-valve to each high-pressure cylinder jacket, which also forms a steam chest containing the two inlet drop valves. The steam passes through the ends of the cylinders, and out through the exhaust drop valves to a pipe which passes it to the centre of the low-pressure cylinders. These are also jacketed by the incoming steam and distribute the steam in a similar manner to the high-pressure cylinders. Each low-pressure cylinder is provided with a jet condenser placed immediately below it. The air pumps for these condensers are 18 in. in diameter by 18-in. stroke, and are driven by pivoted rocking levers from the two cross-heads of the main engines.

The drop valves are of the balanced double mushroom type and are spring loaded. They are worked by eccentrics mounted on a counter-shaft which is driven from the crank shaft through a mitre gear. The cut-off on the high-pressure cylinders can be varied from 0 to $\frac{3}{4}$ stroke by means of the governor gear. On the low-pressure cylinders it can be adjusted by hand between $\frac{1}{2}$ and $\frac{5}{8}$ stroke.

The piston rods in the high-pressure cylinders are $4\frac{1}{2}$ in. in diameter, and in the low-pressure cylinders $5\frac{3}{4}$ in. They are supported between the cylinders on adjustable carriers, lined with white metal. The cross-heads are of cast steel with cast-iron slides. The cross-heads and crank-pin bearings are also white-metal lined, and so arranged that when adjusted for wear the length of the connecting rods is not altered. The flywheel is also a rope drum, a common practice with this type of engine. It is 19 ft. 6 in. in diameter, and is made in two separate pulleys side by side. It carries 34 cotton ropes, each $1\frac{5}{8}$ in. mean diameter. The mean peripheral speed is $\pi DN = \pi \times 19\frac{1}{2} \times 85 = 5200$ ft. per minute, at which speed each rope could transmit $43\frac{1}{2}$ H.P., or a total of $43\frac{1}{2} \times 34 = 1480$ H.P. with safety.

The Sulzer Horizontal Uniflow Steam Engine.—Fig. 66 shows a cross-sectional elevation of a horizontal uniflow steam engine, which has been kindly supplied by the firm of Sulzer Bros., Ltd. The engine has a single cylinder which develops about 400 I.H.P. when running at 150 R.P.M. This cylinder has a diameter of 600 mm. ($23\frac{5}{8}$ in.) and a stroke of 725 mm. ($27\frac{9}{16}$ in.). The cylinder walls are not jacketed, but the covers form the steam chest containing the inlet valves so that the cylinder ends are heated. The steam enters through double-seat spring-loaded mushroom drop valves and exhausts through ports in the centre of the cylinder which are uncovered by the piston itself. These ports are surrounded by an annular chamber which conveys the steam to a jet condenser placed directly below the centre of the cylinder. The air pump is a reversed uniflow, the condensate entering in the centre and being discharged to the hot well through non-return valves at each end in turn. This air pump is driven through a bell-crank lever, and a second connecting rod which works off a small auxiliary crank (not shown) on one end of the main crank shaft.

Supplementary clearance spaces are arranged in the cylinder covers to keep the compression normal when the engine is being started up or is to run non-condensing. At each end of the cylinder a relief valve is fitted, which also communicates with this supplementary clearance space.

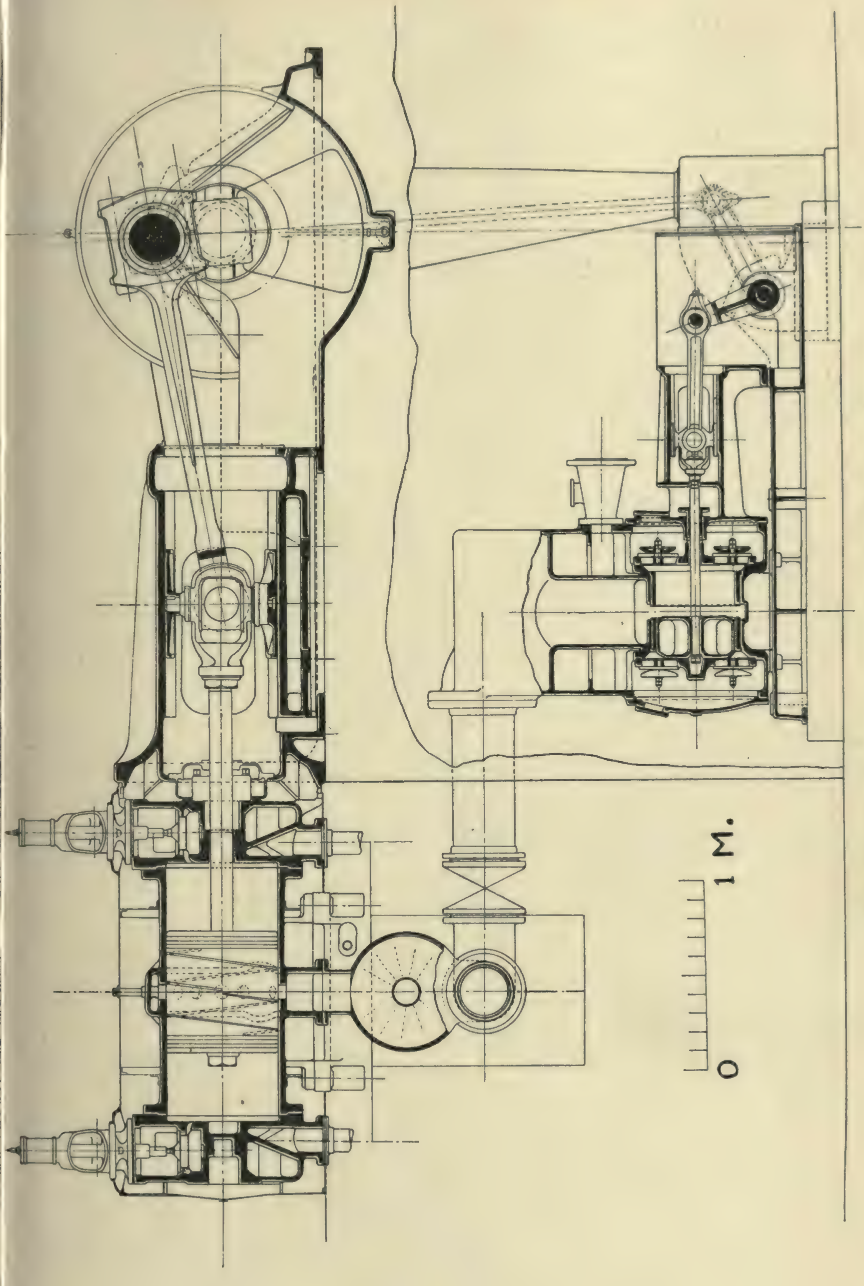


FIG. 66.—Uniflow steam engine (Sulzer) with uniflow air-pump.

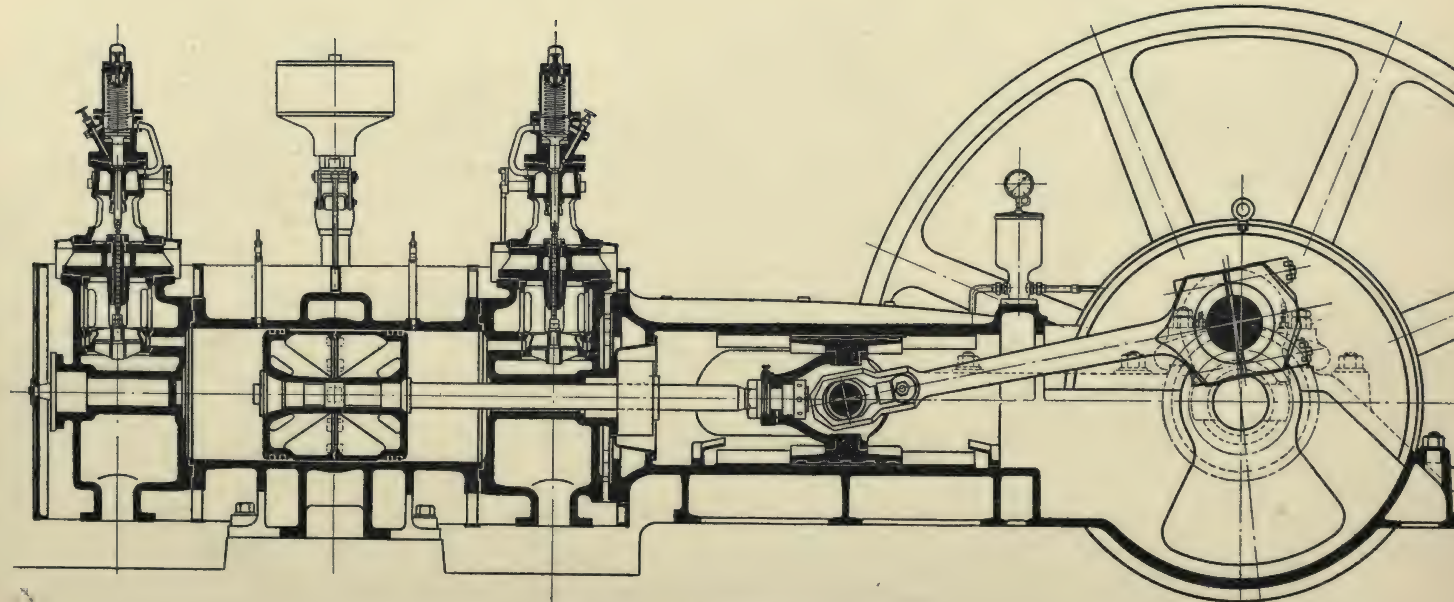


FIG. 67.—Uniflow steam engine (Galloway).

The Galloway Uniflow Steam Engine.—A modern British-built uniflow can be seen in Fig. 67, which is reproduced from the *Engineer* by the courtesy of the Editor.[1] It is made by the firm of Galloways, Ltd., of Manchester, and develops 200 B.H.P. when running at 160 R.P.M. The cylinder is 20 in. in diameter by 24-in. stroke, and is designed to give this power with 160 lb. sq. in. (gauge) on the stop-valve and 120° F. of superheat.

The inlet valves are lifted by oil pressure obtained from small plungers, which are worked by eccentrics off the lay-shaft. The duration of this pressure is controlled by the position of the governor. Upon release the valves are returned to their seats rapidly and quickly by the action of springs and dashpots.

At each end of the cylinder a compression release valve (not shown) is fitted, which in this engine is automatically controlled by the amount of vacuum in the condenser. These valves are opened inwards against springs by means of taper-sleeve cams mounted on the lay-shaft, and the cams are made to slide axially by a small spring-balanced piston, one side of which is open to the atmosphere and the other to the condenser. Under normal running conditions these cams are idle, but for starting up without a vacuum, or should the vacuum fall away, the small piston moves and thus delays the start of the compression in the main cylinder. Sectional drawings of these valve-gears may be found in the *Engineer* article. The guaranteed consumption of this engine is 12·6 lb. of steam per B.H.P. per hour.

The Vertical High-speed Steam Engine

The firm of Belliss & Morcom, Ltd., of Birmingham, make simple, compound, and triple-expansion vertical high-speed engines in standard sizes between 10 B.H.P. and 2500 B.H.P. Their largest size has cylinders 26 in., 36½ in., and 55 in. diameter by 33-in. common stroke, and runs at 187 R.P.M.

A sectional diagram through one type of two-cylinder compound vertical steam engine is reproduced in Fig. 68 by the courtesy of this firm.

Steam enters through a governor-controlled throttle valve and passes round the high-pressure cylinder to a common valve chest placed between the two cylinders. This steam chest contains two piston valves mounted on one rod and driven together by a single eccentric. Whilst the lower valve is passing steam into the high-pressure cylinder (up to the point of cut-off) the upper valve is adjusted to allow the exhaust steam from the high-pressure cylinder to pass inside the valves and direct into the low-pressure cylinder. Steam is thus admitted to the top of one piston and the underneath side of the other simultaneously, and as the cranks are set at 180° to one another the reciprocating parts are better balanced than would otherwise be the case. The paths of the steam can be clearly followed by the arrows on the diagram.

In order to deal with overloads up to 25 per cent. in excess of the normal, this firm have introduced a variation of the above engine in which the piston valves for the high- and low-pressure cylinders are

[1] The *Engineer*, vol. cxxxii. (1921), p. 222.

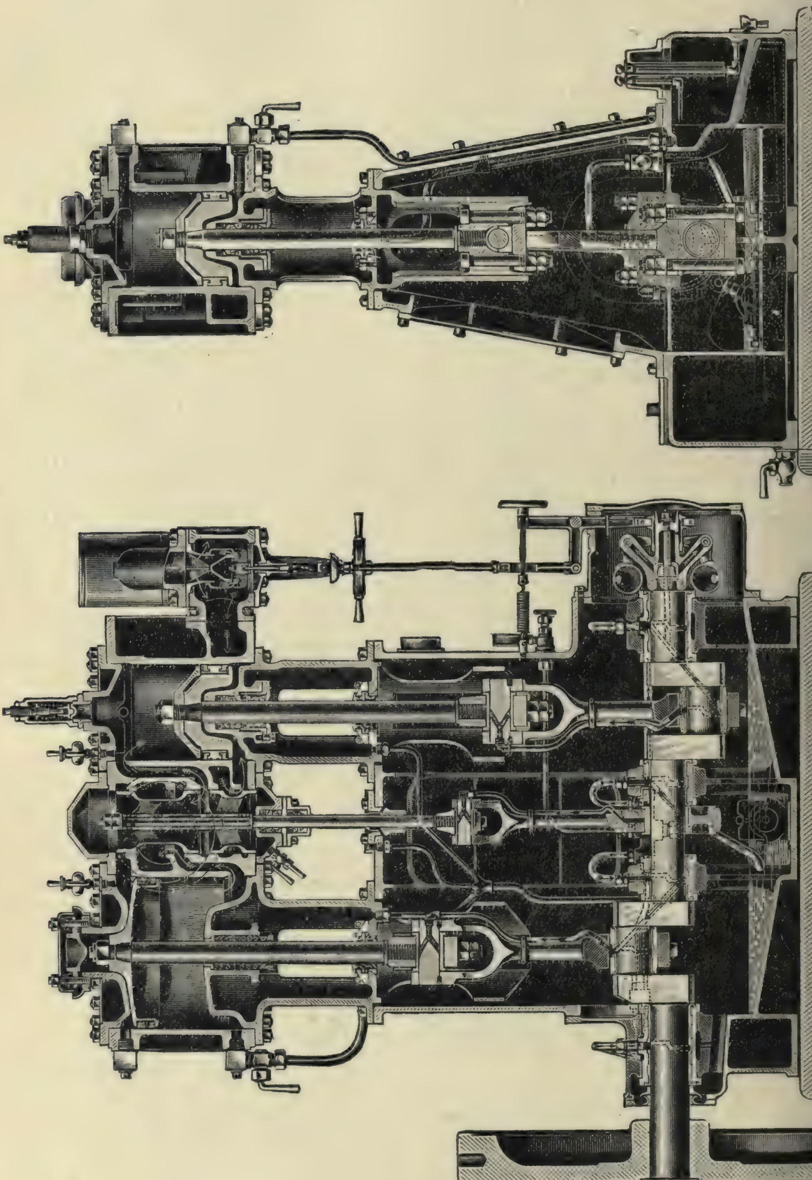


FIG. 68.—Compound vertical high-speed engine with single eccentric (Belliss & Morcom).

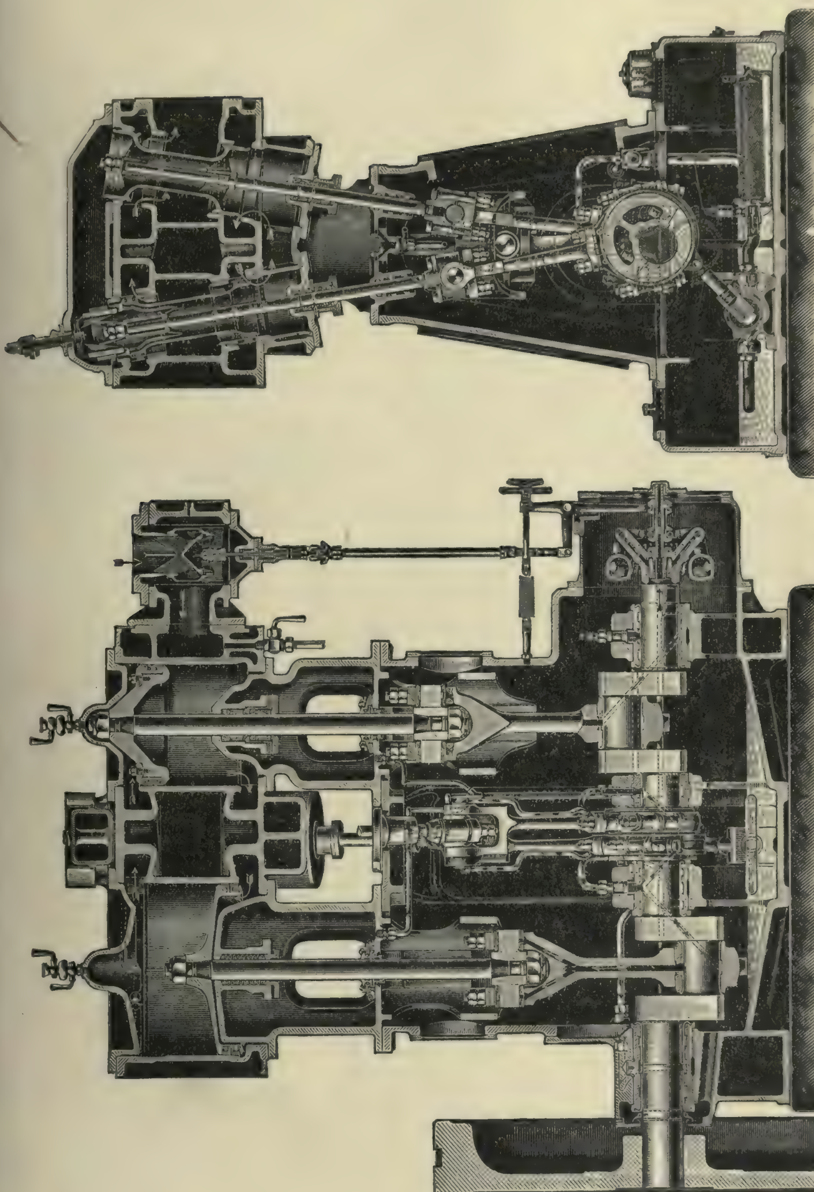


FIG. 69.—Compound vertical high-speed steam engine with V-type valves (Belliss & Morcom).

placed in separate valve chests and an expansion gear fitted to the high-pressure valve. This type of engine can be seen in Fig. 69, also kindly supplied by Messrs. Belliss & Morcom, Ltd., the two valves being set

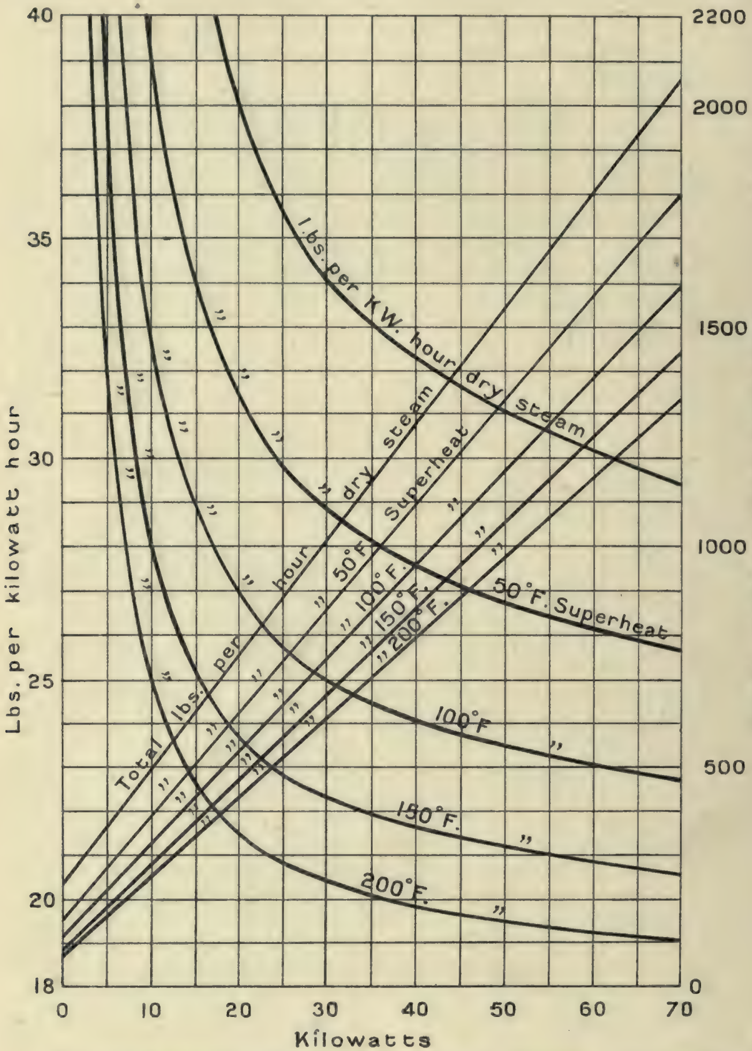


FIG. 70.—Superheat trials on 70 K.W. "Belliss" compound steam engine.

at an angle between the two cylinders (which are vertical) and driven radially from the shaft as a centre by two eccentrics. The high-pressure valve is shown on the left, and the path of the steam is marked with arrows. If this is traced out it will be found to follow the same course as the engine just described. The expansion gear can be varied

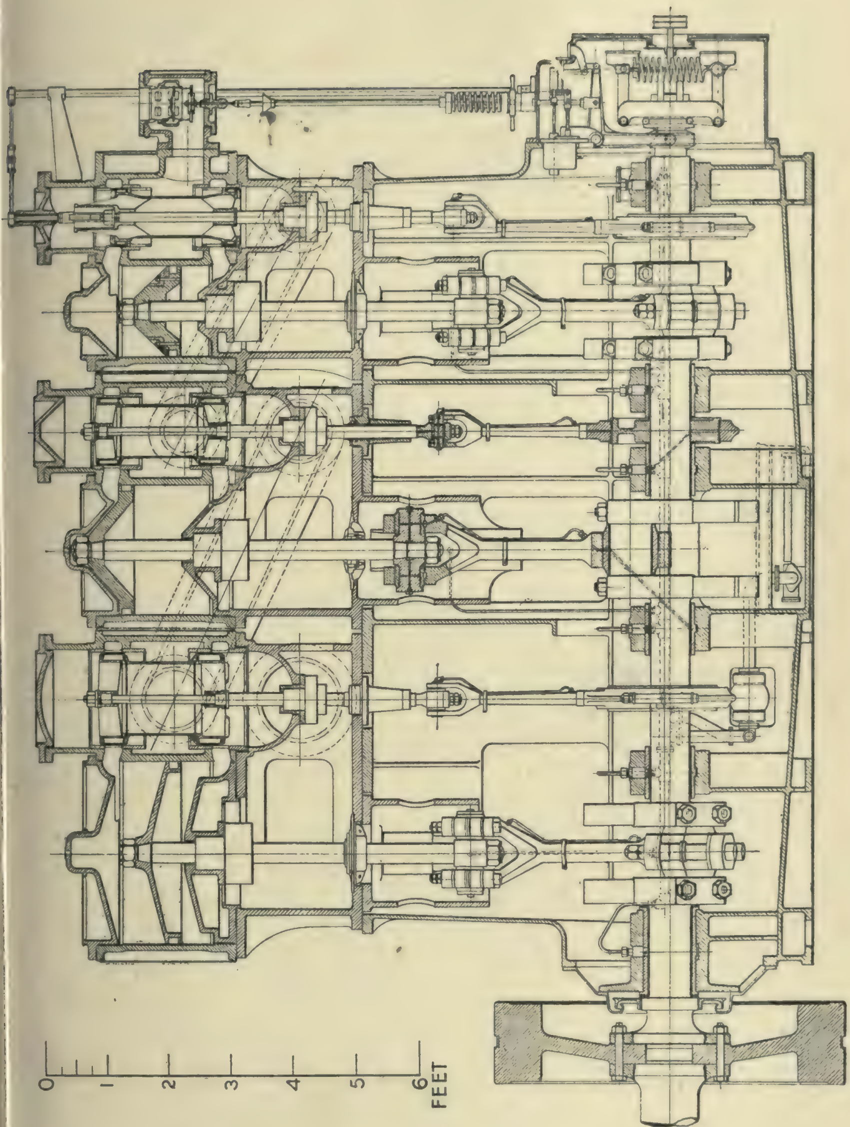


FIG. 71.—Triple-expansion high-speed vertical steam engine (Browett, Lindley).

either by hand or from the governor. The results of some superheat trials on a 70-K.W. set of this type carried out jointly by the University of Birmingham and by representatives of the firm during September and November 1913, are shown in Fig. 70.[1]

The following test results on one of the triple-expansion engines are of interest in showing to what high degrees of superheat it is possible to work with careful design and attention to detail.

TESTS ON 750-K.W. "BELLISS" TRIPLE-EXPANSION ENGINE USING SUPERHEATED STEAM

R.P.M.	230		
Load	$\frac{3}{4}$	full	full
Steam pressure	168	181	180 lb. per sq. in. (gauge).
Total steam temp.	642	606	567° F.
Degrees superheat	268	226	188° F.
Vacuum	26·4	26·2	26·3 in. (barometer 30 in.)
B.H.P.	804	1071	1079
K.W.	562	751	755
Consumption	10·35	10·95	11·45 lb. per B.H.P. per hour.
Assuming a dynamo efficiency of	93·5	94·0	94·0 per cent.
Consumption	14·82	15·6	16·3 lb. per K.W. per hour.

A good example of a modern type of triple-expansion vertical steam engine is shown in Fig. 71 by the courtesy of the makers, Messrs. Browett, Lindley & Co., Ltd., of Patricroft, near Manchester. The engine, which is designed to run direct-coupled to a dynamo, develops 500 K.W. at 330 R.P.M. Each of the three cylinders has the piston type of slide valve driven by eccentric from the main crank shaft. The high-pressure cylinder has in addition an expansion gear, which can vary the cut-off from 0·5 to 0·75 according to the load. The cut-off in the intermediate cylinder is 0·64, and in the low-pressure cylinder 0·68 under normal valve setting. The diameters of the three cylinders are H.P. 16½ in., I.P. 25 in., and L.P. 35 in., and the common stroke is 15 in.

The engine has throttle governing, as can be seen on the right-hand side of the section, and forced lubrication is employed throughout.

[1] See also pp. 169 and 171.

CHAPTER VII

THE PRINCIPLES OF STEAM TURBINE DESIGN

UNLIKE the steam boiler, and to a less extent the steam engine, theoretical considerations are much more directly applicable to the practical design of the steam turbine. It is always necessary to make allowances for mechanical losses and other factors, particularly as our knowledge of the varying conditions of the steam as it passes through a turbine cannot as yet be said to be fully established; but the broad fact remains that the theory of the steam turbine, based on the adiabatic expansion of steam, gives a convenient approximation to results achieved in practice. Familiarity with the methods of applied thermodynamics and with the properties of steam according to the latest authorities is therefore of considerable value to the steam turbine designer.

Turbine Efficiency and Velocity Ratio.—As stated on p. 98, the overall efficiency of a steam turbine is calculated for test purposes as

$$\frac{\text{actual work done on turbine shaft}}{\text{adiabatic heat drop} \times 778}$$

This can equally well be stated as

$$\frac{\text{B.Th.U. converted into work}}{\text{adiabatic B.Th.U. available}} \text{ per lb. of steam.}$$

When first considering a new design it is convenient to neglect friction, windage, and leakage losses, and work on the blading efficiency, or the internal thermal efficiency of the turbine.

The theoretical blading efficiency may be derived as follows :—

The work done per lb. of steam flowing per second $w = \omega T$

where ω = the angular velocity of the blading in radians per second
and T = the torque in ft.-lbs.

Let u = the mean peripheral velocity of the blading in ft./sec.
c = the velocity of the steam leaving the nozzle, that is, entering the moving blade in the direction of the axis of the nozzle, in ft./sec.
v = the resolved component of c in the direction of rotation of the blading.

Similarly, let c_0 = the velocity of exit from the blade in ft./sec.
and v_0 = the resolved component of c_0 as before;
also let r = the mean radius of the blades in ft.
and g = the acceleration due to gravity in ft./sec^2,

v and v_o can be combined vectorially to form V if the velocity diagram of the blade is available. See Fig. 72.

The force on the blades per lb. of steam

$$T = \frac{Vr}{g}$$

for axial flow turbine.[1]

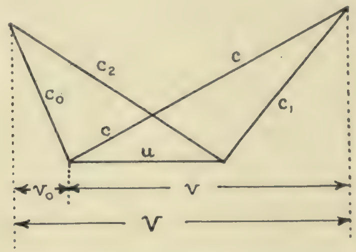


FIG. 72.—Vector diagram for turbine blading.

And the work done per lb. of steam per second

$$w = \frac{\omega Vr}{g}$$

But $$\omega = \frac{u}{r}$$

$$\text{therefore the work done} = \frac{uVr}{rg} = \frac{uV}{g}$$

But the velocity energy available in the steam as it enters the blade

$$= \frac{c^2}{2g}$$

Hence the theoretical thermal efficiency of the blading

$$\eta = \frac{\dfrac{uV}{g}}{\dfrac{c^2}{2g}} = \frac{2uV}{c^2}$$

If there is more than one row of moving blades in one stage, the values of V must be obtained for each row and added together.

Thus, for a 2-row impulse wheel

$$\eta = \frac{2u(V_1 + V_2)}{c^2}$$

[1] "Steam Turbines," by Goudie (1917), p. 161.

This method of arriving at blading efficiency forms a convenient check on blade design, but it must be borne in mind that its value as determined in this way does not take into account the effects of "carry over" friction of the steam, or reheat.

The relation between the blade speed and steam speed, or the value $\frac{u}{c}$, is known as the *velocity ratio* of the turbine, and plays an important part in the design of any particular machine.

The blade speed u may be calculated from the dimensions of the turbine.

$$u = \frac{Rd\pi}{60 \times 12} = \frac{Rd}{230} \text{ ft./sec.}$$

where R = R.P.M.

d = mean diameter of blade ring in in.

The velocity of the steam c is always calculated on the adiabatic heat drop from the relation

$$c^2 = 2gw$$
$$= 2 \times 32{\cdot}2 \times 778 \times I$$

so that $$c = 224\sqrt{I}$$

where I is the adiabatic heat drop between the steam pressures in lb. per sq. in. absolute at the entrance and the exit of the stage.

Fig. 73 shows the relation between the actual thermal efficiency of the blading and the velocity ratio for different types of turbine stages. It should be noted that the efficiencies upon which these curves are based, whilst not including mechanical friction, windage, and leakage losses in the turbine itself, do allow for the frictional effect of the steam through the nozzles and the blading.

It is advisable to keep the velocity ratio of any particular type below the maximum efficiency shown by these curves, partly because the higher velocity ratio generally means a heavier and more costly machine, and partly because the skin friction of the wheels and other losses in the turbine tend to make a lower velocity ratio more efficient. The following range of values is frequently used in modern practice for large turbines :—

	Velocity ratio, *i.e.* $\frac{\text{Blade speed}}{\text{Steam speed}}$
Axial flow reaction (Parsons)	0·75–0·90
One-row impulse wheels	0·45–0·52
Two-row ,, ,, ...	0·22–0·28
Three-row ,, ,, ...	0·13–0·15

The elements of a reaction turbine may be looked upon as nozzles, half of which are fixed and half of which are moving. There is therefore only nozzle loss present. In the impulse turbine there are, in addition, the friction and losses due to the steam passing through the blades. Thus the frictional and other losses are larger in the impulse than in the reaction turbine.

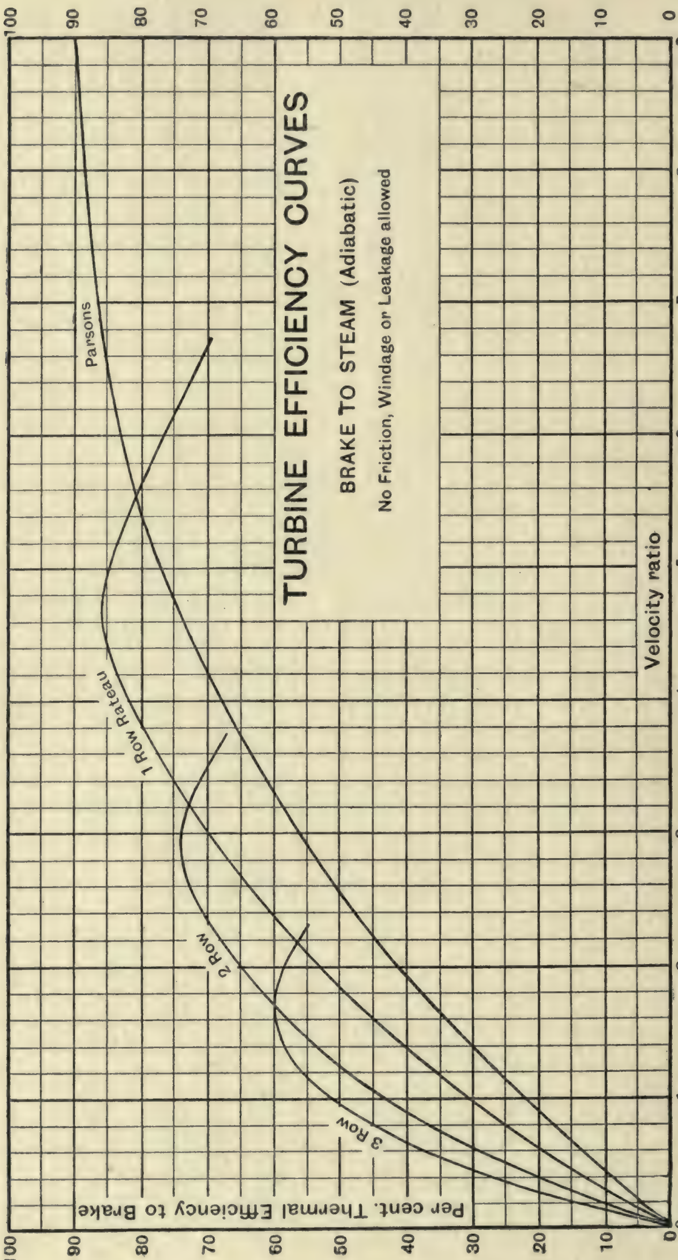


FIG. 73.—Professor Stoney's turbine efficiency curves.

Rotational stresses set up in the discs and blades of impulse turbines limit their mean blade speed (u) to the neighbourhood of 600 ft./sec., though in some cases up to 800 ft./sec. has been allowed, and De Laval, with his solid disc, runs up to 1300 ft./sec.

The design coefficient K and similar turbines.—

Let I = the total adiabatic heat drop through the turbine in B.Th.U.
N = the number of stages reduced to a constant velocity ratio, *i.e.* the number of equivalent one-row wheels in an impulse turbine or the number of rows of moving blades in a reaction turbine
d = the mean diameter of the blading in in.
$\alpha = \frac{u}{c}$ the velocity ratio
R = R.P.M.

then the average steam velocity per stage in an impulse turbine

$$c = 224\sqrt{\frac{I}{N}}$$

But
$$c = \frac{u}{\alpha} \text{ and } u = \frac{Rd}{230}$$

therefore
$$\frac{Rd}{230\alpha} = 224\sqrt{\frac{I}{N}}$$

or
$$\frac{NR^2d^2}{230^2\alpha^2} = 224^2 I$$

or
$$\frac{NR^2d^2 \times 10^{-9}}{\alpha^2} = 2{\cdot}65 I$$

Let $NR^2d^2 \times 10^{-9} = K$,

then
$$\alpha = \frac{u}{c} = \sqrt{\frac{K}{2{\cdot}65 I}}$$

For a reaction turbine this becomes

$$\alpha = \frac{u}{c} = \sqrt{\frac{K}{1{\cdot}32 I}}$$

where there are two rows of working blades in each stage.

The value K forms a convenient coefficient for use in turbine design.

For turbines in which the blade rings are not all the same diameter, K is computed separately for each diameter and added together to obtain the design coefficient,

$$K = [N_1 d_1^2 + N_2 d_2^2 + \text{etc.}]R^2 \times 10^{-9}$$

and from this the mean velocity ratio can he obtained.

N is equal to the number of stages. In the reaction turbine this is taken to mean either the rows of blades on the spindle or in the casing, but not both.

Impulse turbines often consist of a multiple-row wheel followed by

single-row wheels. In that case, since N varies as the square of the velocity ratio, a 2-row wheel is approximately equivalent to $\left(\frac{0\cdot45}{0\cdot26}\right)^2$ = three 1-row wheels. Similarly a 3-row wheel corresponds approximately to $\left(\frac{0\cdot45}{0\cdot15}\right)^2$ = nine 1-row wheels. Similar turbines may be said to be those of different sizes and output but of the same design, that is to say, with the same velocity ratio, stresses, and design coefficient, and the same number of stages.

In such a case, from the definition of the design coefficient K, the diameter varies inversely as the speed, and the area of the steam passages through the blades as the square of the diameter. The total quantity of steam passing through, and therefore approximately the output, varies directly as the square of the diameter or inversely as the square of the speed. For example, taking as a normal machine 3000 K.W. at 3000 R.P.M., then a "similar" machine running at half the speed would give $2^2 = 4$ times the output, or 12,000 K.W.; and one rnning at 1000 R.P.M., 27,000.

Reheat Factor.—As the steam passes through a turbine there is always a certain amount of leakage over the tips of the blades in a reaction type and through the diaphragm glands in an impulse type. In both cases also heat is generated by the friction and eddying of the moving steam, some of which undoubtedly becomes reabsorbed, with the result that the steam in the exhaust tends to be dryer than would be accounted for by pure adiabatic expansion. For both these reasons the specific volume of the steam may be larger and the total heat drop available may be greater than those calculated by the use of adiabatic heat tables. The steam is said to be reheated, and the ratio of the heat drop available to the adiabatic heat drop is known as the "Reheat factor."

The effect of reheating may be seen on the accompanying Mollier diagram (Fig. 74). If the initial state of the steam is represented by the point a_1 on the line of constant pressure p_1, the adiabatic heat drop available in the first stage of a turbine in which the steam expands to a pressure p_2 is shown by the vertical line a_1a_2. If frictional losses are taken into account the (adiabatic) heat drop will be

$$\eta_s \times a_1a_2 = a_1c_2$$

where η_s is the stage efficiency based on such losses. The reheating effect causes the point c_2 to move at constant heat to b_2 on the pressure line p_2, and this represents the condition of the steam at the end of the first stage. If the process is repeated for a multi-stage turbine working between p_1 and p_0 a series of steps are obtained, and since the lines of constant pressure diverge on a Mollier diagram it is evident that the available heat drop, $a_1a_2 + b_2a_3 + b_3a_4 + b_4a_0$, is slightly more than the adiabatic heat drop as represented by the line a_1d_0. As already stated, the ratio between these heat drops gives the reheat factor. Provided the stage efficiencies are known or assumed this method can be applied, whether the steam is originally superheated or whether the steps cross the saturation line or no. In the latter case it is probable that a more accurate forecast of the reheat factor would be obtained if the pressure

curves for some distance below the saturation line represented a supersaturated condition of the steam rather than a state of thermal equilibrium. Such an Iϕ-chart, based on Callendar's tables, has been computed and drawn by H. M. Martin.[1] The effect of supersaturation is indicated on Fig. 74 by the dotted line.

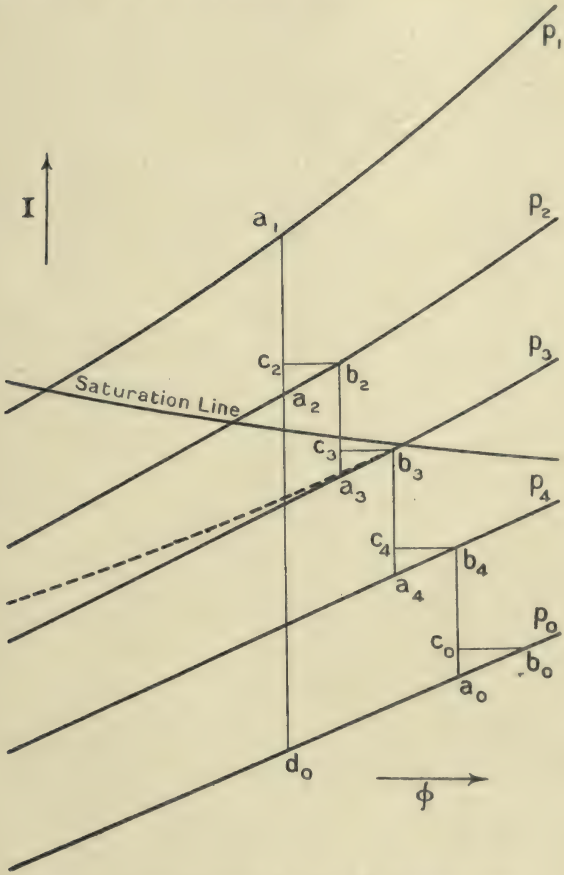


FIG. 74.—"Reheat" effect shown on Mollier diagram.

Quantity of Steam passing through Nozzles.—In order to arrive at suitable proportions for the cross-sectional areas of nozzles and passages through the blading of a steam turbine, it is convenient to be able to estimate quickly the amount of steam that can flow under any given conditions. All books on applied thermodynamics show that this discharge reaches a maximum when the drop in pressure is more than a certain *critical* value. For initially dry saturated steam this point is reached when the final pressure is about 0·58 times the initial pressure.

[1] "A New Theory of the Steam Turbine," *Engineering*, vol. cvi. p. 1.

Any further drop in pressure will not increase the quantity of steam that can be passed. It can also be shown that the velocity of steam (c) at the critical is equal to the velocity of sound in saturated steam, say, 1480 ft./sec. If, therefore, from the relation $c = 224\sqrt{I}$, the velocity of the steam is found to be greater than 1480 ft./sec., the drop in pressure is above the critical and formulæ for the maximum discharge should be used. Such formulæ can be written in the form—

$$Q = Map \text{ lb./hr.*}$$

where Q = quantity of steam in lb./hr.
a = area of nozzle in sq. in.
p = the initial pressure in lb. per sq. in. (abs.).
and M has the following values for initially dry saturated steam :—

Initial pressure lb. per sq. in. (abs.).	Value of M in equation $Q = Map$ (Rateau).
300	50·7
250	51·0
200	51·4
150	51·8
100	52·5
50	53·5
14·7	55·4
10	56·0
5	57·2
1·0	59·7

These values may be corrected for superheat in the proportion of

$$\frac{\sqrt{\text{abs. temp. saturated}}}{\sqrt{\text{abs. temp. superheated}}}$$

e.g. at 250 lb. per sq. in. (abs.) with 200° F. superheat

$$M_s = 51{\cdot}0\sqrt{\frac{862}{1062}} = 46$$

On the other hand, if the critical has not been reached, the equation of continuity gives the following relation :—

$$Ac = qV$$

where A = area in sq. ft.
c = velocity of flow in ft./sec.
q = quantity of steam in lb./sec.
V = volume of steam in cub. ft./lb. at exit from the nozzle.

In practice the following dimensional units are more convenient :—

a = area of nozzle in sq. in. = 144A.
c = velocity of flow in ft./sec.
Q = quantity of steam in lb./hr. = $3600q$.
V = volume of steam in cub. ft./lb.

* See article on "The Physics of the Steam Turbine," in "The Dictionary of Applied Physics."

Then
$$\frac{a}{144} \cdot c = \frac{Q}{3600} \cdot V$$
or
$$Q = \frac{25 \cdot a \cdot c}{V} \text{ lb./hr.}$$

which is a convenient formula to use when the pressure drop (or the velocity) is less than the critical values.

Since it is customary to aim at keeping the velocity ratio $\left(\frac{u}{c}\right)$ constant in any particular turbine or part of a turbine, it follows that $\frac{q}{c} = \frac{A}{V} =$ a constant, when the blade speed and the blade diameter are kept constant. From this the area of the steam passages must be increased in proportion to the increasing specific volume of the steam as the pressure drops. This is one of the limiting factors in the design of large turbines.

Application to the Provisional Design of an Impulse Turbine.—In designing any particular turbine, the following conditions would be known, and will be assumed in this case at the values given.

Type	Impulse—2-row wheel followed by 1-row wheels.
Output	3000 K.W.
Speed	3000 R.P.M.
Initial steam pressure on boiler side of stop valve	200 lb. per sq. in. (gauge).
Superheat	210° F.
Vacuum	29 in. Hg (referred to a 30-in. barometer).

If the blade speed (u) is limited by rotational stresses to 600 ft./sec., then the mean diameter of the blade ring (d) is determined directly from the relation—

$$u = \frac{Rd}{230} \quad \text{or } 600 = \frac{3000d}{230}$$
whence
$$d = 46 \text{ in.}$$

To estimate the number of wheels the available heat drop through the turbine is required. This can be considered in various ways, but for the moment simple adiabatic expansion will be assumed. By using Peabody's temperature-entropy tables,[1] the adiabatic B.Th.U. available may be obtained as follows :—

215 lb. per sq. in. (abs.) 210° F. superheat	$\phi = 1{\cdot}663$	B.Th.U. =	1315
29 in. = 0·5 lb. per sq. in. (abs.) at	$\phi = 1{\cdot}663$	B.Th.U. =	895
Adiabatic B.Th.U. available (by difference)			420

[1] Eighth edition, 1914 (London : Chapman & Hall).

As Peabody's tables only go down to 0·594 lb. per sq. in., it is necessary here to exterpolate down to 0·5 lb. per sq. in., but the heat drop to temperature curve may be taken as a straight line in this region, which enables the required figures to be easily obtained.

Velocity ratios $\left(\frac{u}{c}\right)$ have now to be chosen with the aid of the curves in Fig. 73 or the table on p. 133.

Assume $$\frac{u}{c} = 0·26 \text{ for 2-row wheel}$$
$$= 0·45 \text{ for 1-row wheels}$$

Then for the 2-row wheel

$$c = \frac{600}{0·26} = 2310 \text{ ft./sec.}$$

and the adiabatic heat drop in the 2-row wheel is obtained from the relation $c = 224\sqrt{\text{I}}$

$$\text{I} = \frac{2310^2}{224^2} = 106 \text{ B.Th.U.}$$

or $\frac{106}{420} \times 100 = 25$ per cent. of the total heat drop available.

This is quite a common percentage for this type of turbine, in which it may vary between 20 and 30 per cent.

Similarly, in the 1-row wheels

$$c = \frac{600}{0·45} = 1330 \text{ ft./sec.}$$

and the adiabatic heat drop in each 1-row wheel

$$= \frac{1330^2}{224^2} = 35 \text{ B.Th.U.}$$

Since the B.Th.U. available[1] in all the 1-row wheels is 420 − 106 = 314, the number of 1-row wheels required will be

$$\tfrac{314}{35} = 9$$

The turbine will therefore have one 2-row wheel followed by nine 1-row wheels, all 46 in. mean blade ring diameter.

The design coefficient (K)

$$\text{K} = \text{N}d^2\text{R}^2 10^{-9}$$

where N = 12 equivalent 1-row wheels (see p. 136).

[1] As will be seen in the next paragraph, 420 B.Th.U. are not strictly available for the nozzles and blading of the turbine. The loss through the governor gear and stop valve has to be accounted for on the one side, and the reheating effect on the other.

In this case—

$$K = 12 \times 46^2 \times 3000^2 \times 10^{-9}$$
$$= 228$$

But
$$\frac{u}{c} = \sqrt{\frac{K}{2{\cdot}65 \times I}}$$
$$= \sqrt{\frac{228}{2{\cdot}65 \times 420}}$$
$$= 0{\cdot}453$$

which forms a check on the velocity ratio of 0·45 provisionally assumed for the 1-row wheels.

Proportions of Nozzles.—If it is desired to proceed further with the design of such a turbine, and calculate the proportions of the various nozzles, it is advisable to modify the above method of arriving at the heat drop, so as to conform more with conditions pertaining in actual practice.

The initial steam pressure and temperature are measured on the boiler side of the stop valve. As the steam passes through the valve and the governor gear there will be a drop in pressure before it reaches the first nozzles of the turbine. This is frequently assumed to be 15 lb. per sq. in. The total temperature, on the other hand, will remain the same, except for one or two degrees radiation loss.

At 215 lb. per sq. in. (abs.) the temperature of saturation is 388° F., which with 210° F. superheat gives a total temperature of 598° F. Allowing 2° F. for radiation loss leaves the total temperature of the steam entering the first nozzles at 596° F.

But the pressure has dropped to 200 lb. per sq. in., at which the temperature of saturation is 382° F., leaving 596 − 382 = 214° F. superheat. As before, the adiabatic B.Th.U. available in the turbine is then

200 lb. per sq. in. (abs.) 214° F. superheat	$\phi = 1{\cdot}67$	B.Th.U.	= 1315
0·5 lb. per sq. in. (abs.) at	$\phi = 1{\cdot}67$	"	= 899
Adiabatic B.Th.U. available (by difference)			= 416

This shows a loss of 4 B.Th.U. or 1 per cent. in the stop valve and governor gear, which is rather excessive, but on the safe side for purposes of design.

The heat drop in the 2-row wheel will be 106 B.Th.U. as before (p. 140), so that there will now be 416 − 106 = 310 B.Th.U., or $\frac{310}{9} = 34{\cdot}4$ B.Th.U. available per stage in the nine 1-row wheels. This can conveniently be taken in the design as 34 and 35 B.Th.U. in each alternate wheel.

Owing to the causes pointed out in the paragraph on the reheat factor (p. 136) the actual B.Th.U. through the turbine will be higher than 416, but as correct values of the reheat factor cannot as yet be considered available, this adjustment is better left to the discretion of the individual designer. It will be quite near enough for the purposes of this design

if the reheat factor is omitted in estimating the pressure drop, and an approximation made in the probable increase in the volume of the exhaust steam due to reheating. This may be arrived at as follows: The adiabatic specific volume of the steam at 0·5 lb. per sq. in. and $\phi = 1·67$ is found from the tables to be 526 cub. ft. This corresponds to a heat drop of 416 B.Th.U.

If now an internal efficiency for the turbine is estimated from the curves in Fig. 73 it will actually be somewhere between the 2-row and the 1-row wheels, say 82 per cent. Then the B.Th.U. actually used in the turbine will be $416 \times 0·82 = 341$, and the heat actually thrown away in the exhaust will be $1315 - 341 = 974$ B.Th.U.

From the tables 974 B.Th.U. at 0·5 lb. per sq. in. corresponds to an entropy of 1·81 (by exterpolation), and a specific volume of 573 cub. ft. This is an estimated increase in specific volume of $573 - 526 = 47$ cub. ft. or 9·0 per cent. at the exhaust end after the steam had left the last row of blades. In the absence of more definite information this increase may be divided evenly over the nine 1-row wheels of the turbine. The adiabatic specific volume obtained from the table for the first 1-row wheel should be increased 1 per cent., for the second 1-row wheel 2 per cent., and so on up to 9 per cent. for the ninth or last 1-row wheel.

To make this clearer the design may be tabulated somewhat as follows:—

TABLE FOR 3000 K.W. 3000 R.P.M. IMPULSE TURBINE.

1 Stage No.	2 B.Th.U. per stage.	3 Total adiabatic heat.	4 Pressure in lb./in.² (abs.) at $\phi = 1·67$.	5 Corresponding specific volume. (adiabatic).	6 Add for reheat (per cent.).	7 Final sp. vol. (cu. ft.).	8 Nozzle area (sq. in.).
1	—	1315	200	—	—	—	3·6
	106	1209	68	—	—	—	—
2	35	1174	45	9·4	1	9·5	9·4
3	34	1140	29	14	2	14·3	14·2
4	35	1105	18	21	3	22	21·8
5	34	1071	10·6	34	4	35	34·7
6	35	1036	6·3	53	5	56	55·5
7	34	1002	3·5	89	6	94	93·3
8	35	967	1·9	153	7	164	162·5
9	34	933	1·0	276	8	294	295·5
10	35	898	0·5	526	9	573	568

In this table—

Column 1 sets out the stages of the turbine.

„ 2 is calculated as already shown on pp. 140 and 141.

„ 3 is obtained by subtracting the values in col. 2 from the total B.Th.U. available.

„ 4 and 5 are obtained from steam tables corresponding to the values in col. 3.

„ 6 represents the estimated percentage increase in specific volume due to reheat as calculated on p. 142.

Column 7 gives the corrected volumes used to calculate nozzle areas.

,, 8 may be obtained from col. 7 when the total quantity of steam passing through the turbine is known.

On p. 97 it was shown that the number of lb. of steam required by the ideal turbine per K.W. was $\frac{3412}{\text{B.Th.U.}}$

To estimate the actual quantity used an efficiency for the whole turbine and alternator must be chosen. If this is taken as 74 per cent. and the heat drop (B.Th.U.) as 420, then theoretically the total quantity of steam used in lb. per hour for a 3000-K.W. machine is

$$\frac{3412}{420} \times \frac{1}{0{\cdot}74} \times 3000 = 33{,}000 \text{ lb./hr.}$$

The first nozzle is above the critical (p. 138), so that

$$a = \frac{Q}{M_s p}$$

at 200 lb. per sq. in. and 214° F. superheat.

$$\begin{aligned} M_s &= 51{\cdot}4 \times \sqrt{\frac{382 + 460}{214 + 382 + 460}} \\ &= 51{\cdot}4 \times \sqrt{\frac{842}{1056}} \\ &= 45{\cdot}8 \end{aligned}$$

therefore $$a = \frac{33{,}000}{45{\cdot}8 \times 200} = 3{\cdot}6 \text{ sq. in. for full load.}$$

It should perhaps be made clear that the values of M given on p. 138 are derived from Rateau's formula for the actual discharge of a nozzle above the critical. This formula allows a margin for nozzle loss, which in this case would be about 3 per cent. If values for M are calculated direct from theory, this margin should be allowed for.

The other stages are below the critical, so that

$$a = \frac{QV}{25c} \quad \text{(p. 139)}$$

where $c = 1330$ ft./sec. (p. 140)

$$\begin{aligned} a &= \frac{33{,}000V}{25 \times 1330} \\ &= 0{\cdot}993V \end{aligned}$$

where V = corrected specific volume in cub. ft. shown in col. 7.

The values for a are shown in col. 8.

Another Method of determining the Proportions of the Impulse Turbine.—The conception of the height of an imaginary homogeneous atmosphere, referred to by Deschanel,[1] may conveniently be extended to steam. This homogeneous head

$$H = 144pV$$

where p is the pressure in lb. per sq. in. and V is the specific volume of 1 lb. of steam at that pressure.

For saturated steam the following empirical formula may be used—

$$H = 8000(6 + \log p)$$

For superheated steam, which has nearly the properties of a perfect gas, the homogeneous head may be taken as proportional to the absolute temperature; so that if T is the temperature of saturated steam and T_s the temperature of superheated steam

$$H_s = H\frac{T_s}{T}$$

The use of this homogeneous head in the design of steam turbines (both impulse and reaction) is fully explained in an article by Professor Stoney.[2] It has the advantage of avoiding to a large degree the use of entropy tables or diagrams. It also does not require a knowledge of the distribution of the reheat factor through the various stages of the turbine.

In this method the homogeneous head is assumed to be equally distributed over each stage, and the values are obtained by calculating the two heads for the initial steam conditions and for the exhaust (allowing for reheat) and then filling in the intermediate values on the above assumption.

The work done per stage (w) is also constant, and may be written

$$w = 144\frac{\gamma}{\gamma - 1}(p_1V_1 - p_2V_2) = \frac{\gamma}{\gamma - 1}(H_1 - H_2)$$

now, $\gamma = 1{\cdot}3$ for superheated steam and about $1{\cdot}135$ for saturated steam, so that $\frac{\gamma}{\gamma - 1} = \frac{1{\cdot}3}{0{\cdot}3} = 4{\cdot}3$ for superheat, and $\frac{1{\cdot}135}{0{\cdot}135} = 8{\cdot}4$ for saturated steam. Hence for constant work done per stage the drop of homogeneous head per stage for the superheated portion is practically double that for the saturated part of the turbine.

The point at which this change takes place is difficult to obtain exactly, but it can be estimated with sufficient accuracy by assuming an internal efficiency for the whole turbine and then noting where the condition curve[3] crosses the saturation line on a Mollier diagram.

In order to determine the volume of the steam and therefore the

[1] "Natural Philosophy," Everett's translation, 1875, p. 162.

[2] "Pressure Distribution in Steam Turbines," *Engineering*, vol. cxi. (1921), p. 250. See also p. 293 for errata.

[3] For an explanation of the condition curve of a turbine, see "Steam Turbines," by W. J. Goudie (1917), p. 328 *et seq.*

area of the nozzles the homogeneous head is preferably taken at the exhaust of each stage. It can be conveniently estimated for each stage by a graphical method used in the worked example below.

The fundamental equation for work done (w) when steam expands adiabatically in a stage from p_1 to p_2 may be written—

$$\begin{aligned} w &= \int_{p_2}^{p_1} \mathrm{H}\frac{dp}{p} \\ &= \mathrm{H}(\log_e p_1 - \log_e p_2) \\ &= \mathrm{H} \log_e \left(\frac{p_1}{p_2}\right) \end{aligned}$$

or putting the ratio $\frac{p_1}{p_2} = \mathrm{X}$,

$$\begin{aligned} w &= \mathrm{H} \log_e \mathrm{X} \\ &= 2{\cdot}3\ \mathrm{H} \log_{10} \mathrm{X} \end{aligned}$$

This is assumed to be constant per stage.

Again, $w = 778\Delta\mathrm{I}$, where $\Delta\mathrm{I}$ is the heat drop per stage.

Hence, by equating the two values for w

$$\log \mathrm{X} = \frac{778\Delta\mathrm{I}}{2{\cdot}3\mathrm{H}} = \frac{\mathrm{C}}{\mathrm{H}}$$

where C is a constant.

For the pressure drop through the whole turbine from p_1 to p_0

$$\log \frac{p_1}{p_0} = \frac{778\mathrm{I}}{2{\cdot}3} \sum \frac{1}{\mathrm{H}} = \mathrm{C} \sum \frac{1}{\mathrm{H}}$$

where C is the constant and I the total heat drop through the turbine.

To allow for the effect of reheat, I should be calculated as before (p. 141), and $\log \frac{p_1}{p_0}$ can be obtained direct from the initial and final steam conditions. Alternatively, without having to find the reheat factor, which is uncertain, $\sum \frac{1}{\mathrm{H}}$ can be obtained from a table of reciprocals, and then

$$\mathrm{C} = \frac{\log \frac{p_1}{p_0}}{\sum \frac{1}{\mathrm{H}}}$$

This enables C for the whole turbine to be obtained, and thus log X for each stage from the relation $\frac{\mathrm{C}}{\mathrm{H}}$. As a check on the working the sum of the values obtained for log X should add up to $\log \frac{p_1}{p_0}$.

Again, since $\log p_1 - \log p_2 = \frac{\mathrm{C}}{\mathrm{H}}$, by evaluating $\log p_0$ and adding

L

the result to log X of the last stage the value of the log p at the end of the preceeding stage is at once obtained. A repetition of this process upwards through all the stages of the turbine gives values for the corresponding log p of each stage, and hence the pressure distribution throughout the turbine.

The corresponding specific volumes are directly determined from the relation

$$V = \frac{H}{144p}$$

If the turbine has 2-row or 3-row wheels the equivalent 1-row wheels, as explained in p. 136, should be used to tabulate the equivalent number of 1-row stages. That is, a 2-row wheel is approximately equivalent to three 1-row wheels, and a 3-row wheel to nine 1-row wheels.

It is also advisable, since both H and C run to five figures, to use five- or seven-figure logarithms to be sure of the last significant figure of the four-figure log column when checking across.

Application of the Homogeneous Head Method to the Design of an Impulse Turbine.—It may be of interest to apply this method to the 3000 K.W. 3000 R.P.M. Impulse turbine which has been outlined on p. 142. The steam conditions at the entry to the first nozzle are shown on p. 142 to be 200 lb. per sq. in. (abs.) with 214° F. superheat ($\phi = 1{\cdot}67$). The pressure at the exhaust of the last row of blades is taken as 0·5 lb. per sq. in., but the entropy (see p. 141) has been increased to 1·81, to allow for reheat when an efficiency of 82 per cent. is assumed.

The turbine has one 2-row wheel followed by nine 1-row wheels, so that the equivalent 1-row stages $= 3 + 9 = 12$ (see p. 136).

The upper part of Fig. 75 has been traced from the Mollier chart at the end of Peabody's Steam Tables (8th edit. 1914), and is self-explanatory. It is near enough to assume that the steam condition line is straight, and the point where it crosses the saturation line is an indication of the point where the homogeneous head line (see lower half of Fig. 75) changes slope. In the light of our present knowledge this point is really indeterminate, as the exact point where condensation begins in a turbine is rather doubtful. It may lie anywhere between the saturation line and the "Wilson" line. It should also be noted that considerable variation exists between various steam charts. Peabody's chart is used here only because his temperature-entropy tables were used in the former example.

A graphical construction for the H curves is as follows: to make the slope of the superheated portion twice that of the saturated portion, $\tan\beta$ must equal $2\tan\alpha$. Calculate H and H_0 from the steam conditions. At 200 lb. per sq. in., $\phi = 1{\cdot}67$ and the specific volume $= 3{\cdot}085$ cub. ft. per lb. of steam, hence the homogeneous head at the entry to the first nozzle

$$H = 144 \times 200 \times 3{\cdot}085 = 88{,}800$$

and at the exhaust (from data on p. 142)

$$H_0 = 144 \times 0{\cdot}5 \times 573 = 41{,}250$$

Choose a convenient vertical scale for H, and sub-divide the horizontal

distance AE into 12 equal parts to represent the (assumed) equal distribution of heat drop through the 12 equivalent 1-row stages.

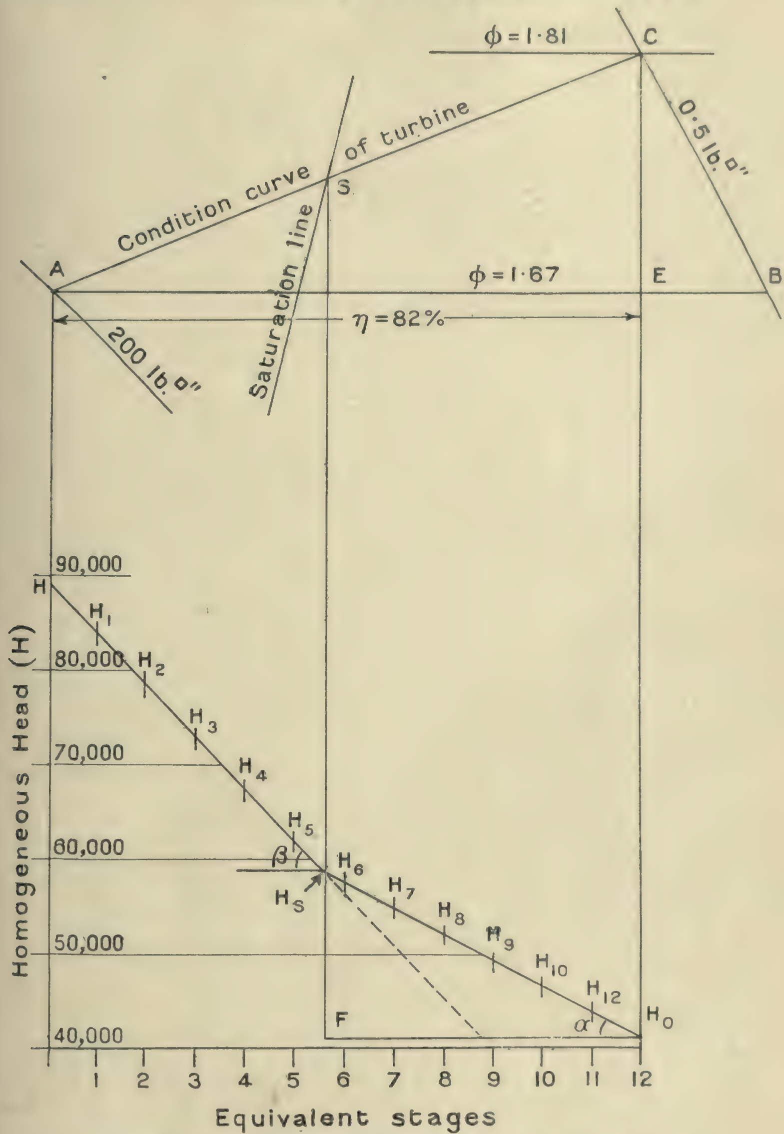


FIG. 75.—Homogeneous head method of steam turbine design.

Divide this horizontal scale in the ratio $\frac{AS}{SC}$ by dropping a perpendicular from S to F. Join H to the mid-point between F and H_0,

cutting this perpendicular in H_8. HH_8 and H_8H_0 are then the required lines from which to read off the intermediate homogeneous heads at the exhaust of each of the 12 stages

TABLE FOR 3000 K.W. 3000 R.P.M. IMPULSE TURBINE, CALCULATED BY THE HOMOGENEOUS HEAD METHOD.

1		2	3	4	5	6	7
Equivalent stage No.		H	$\frac{1}{H} \times 10^5$	$\log X = \frac{C}{H}$	$\log p$	p	$V = \frac{H}{144p}$
	—	88,800	—	—	2·3011	200	—
1.	1	83,800	1·193	0·1456	2·1555	—	—
	2	78,500	1·274	0·1554	2·0001	—	—
	3	72,800	1·374	0·1677	1·8324	68·0	—
2.	4	68,000	1·471	0·1795	1·6529	45·0	10·5
3.	5	61,600	1·623	0·1981	1·4548	28·5	15·0
4.	6	57,400	1·742	0·2126	1·2422	17·5	22·8
5.	7	54,700	1·828	0·2231	1·0191	10·5	36·2
6.	8	52,000	1·923	0·2342	0·7849	6·1	59·2
7.	9	49,100	2·037	0·2486	0·5363	3·4	97·6
8.	10	46,500	2·151	0·2625	0·2738	1·9	170·0
9.	11	43,700	2·288	0·2793	$\bar{1}$·9945	1·0	303·3
10.	12	41,300	2·421	0·2955	$\bar{1}$·6990	0·5	573
			21·325	2·6021			

$$\log \frac{p_1}{p_0} = \log \frac{200}{0·5} = 2·6021 \qquad C = \frac{2·6021}{21·325 \times 10^{-5}} = 12,205$$

These values are tabulated in column 2 of the accompanying table, and the reciprocals added in column 3 from a table of reciprocals.

Now
$$\log \frac{p_1}{p_0} = \log \frac{200}{0·5} = 2·6021$$

and the sum of all the reciprocals in column 3 $= 21·325 = \Sigma\frac{1}{H}$, hence the total constant

$$C = \frac{\log \frac{p_1}{p_0}}{\Sigma \frac{1}{H}} = \frac{2·6021}{21·325 \times 10^{-5}} = 12,205$$

Column 4 is obtained by dividing this constant by the corresponding H. It is better to use five- or seven-figure logarithms for this, as the total, correct to four significant figures, must check with $\log \frac{p_1}{p_0}$. Starting at the bottom of column 5 with $\log p_0 = \log 0·5 = \bar{1}·6990$, and adding the corresponding log X, gives the $\log p$ of the preceding stage thus—

$$\bar{1}·6990 + 0·2955 = \bar{1}·9945$$

Repeating this process all the way up, the top value 2·3011 should, as a check, equal log p_1, *i.e.* 200 lb. per sq. in.

It is then a simple matter to write down the pressures in column 6 from their corresponding logarithmic values, and obtain the specific volumes in column 7 from the relation $V = \frac{H}{144p}$ for each stage. These values should be compared with those obtained by the previous method on p. 142.

The Design of the Axial-Flow Reaction Steam Turbine (Parsons Type).—The reaction turbine has a large number of stages compared with the impulse type, and expansion of the steam takes place partly in the fixed and partly in the moving blades of each stage, instead of practically all in the fixed blades or nozzles, as is the case with the impulse type. For purposes of design, equal expansion in the fixed and moving blades is assumed; and the large number of stages ensures that the pressure drop through each stage is always less than the critical. Since it is not advisable in general to have a greater blade height than $\frac{d}{5}$, where $d =$ the mean diameter of the blade ring, the spindle diameter is stepped up as occasion requires, a common practice being to have three steps, which are then called the high-pressure, intermediate-pressure, and low-pressure cylinders respectively. Spindle diameters are arbitrarily fixed, usual proportions being $1 : \sqrt{2} : 2$ or $1 : \sqrt{3} : 3$.

The ideal turbine would have successively increasing blade heights in each cylinder to form what may be called a "conical" turbine; but in order to make machining practicable, it is often arranged for the blades to form a series of parallel lines, though there may be several such sections in each cylinder.

The fact that expansion takes place in both the fixed and the moving blades, and the large number of stages, make it difficult to use the first heat-drop method outlined for the impulse turbine on p. 139; it is better to calculate the pressure throughout the turbine in the following way, and use the homogeneous head method.[1]

The Pressure along an Axial-Flow Reaction Turbine.—For the sake of simplicity this is calculated for saturated steam. Since the pressure drop in each reaction stage is always less than the critical the equation of continuity gives the relation—

$$c = \frac{QV}{3600A} \quad \text{or} \quad \frac{QV \times 144}{3600a}$$

where $c =$ steam speed in ft./sec.
$Q =$ quantity of steam passing in lb. per hour
$V =$ specific volume of steam
$A =$ area of nozzle in sq. ft.
$a =$ area of nozzle in sq. in.

[1] "Pressure Distribution in Steam Turbines," *Engineering*, vol. cxi. (1921), p. 250.

From the definition of the homogeneous head (H)—

$$V = \frac{H}{144p}$$

so that
$$c = \frac{QH}{3600pa}$$

and
$$c^2 = \frac{Q^2H^2}{12{\cdot}96 \times 10^6 p^2 a^2}$$

Again, for a drop in pressure (dp) in each row of blades, whether on the casing or on the rotor,

$$c^2 = 2g \times 144 V dp$$
$$= 2gH\frac{dp}{p}$$

Equating these two values for c^2 gives

$$pdp = \frac{Q^2H}{2g \times 12{\cdot}96 \times 10^6 \times a^2}$$

Integrating between p_1 and p_2, and including 2N, the number of stages—

$$\tfrac{1}{2}(p_1{}^2 - p_2{}^2) = \frac{2NQ^2H}{2g \times 12{\cdot}96 \times 10^6 \times a^2}\ *$$

whence
$$p_1{}^2 - p_2{}^2 = \frac{NQ^2H}{209 \times 10^6 \times a^2}$$

For the purposes of applying the homogeneous head method to the design of the reaction turbine it simplifies the calculation to treat a number of rows of blades which are of uniform height and diameter as a single section, and assume that the heat drop through each section is approximately the same. The above equation will then give a relation between the pressures p_1 and p_2 at each end of such a section.

In practice it is more convenient to work with the mean ring diameter (d) and the blade height (h) than with the area (a) of the nozzle. The nominal height is the difference of the radii of the casing and the rotor.

The nozzle area will equal

$$a = \pi\beta d\left(h + \delta\frac{1 - \beta}{\beta}\right)$$

where β is the ratio of the blade opening to the blade pitch, $= 0{\cdot}35$ for normal Parsons blading, $\delta =$ the clearance between the tip of the blade and the casing.

* A more exact formula is given by H. M. Martin in *Engineering*, Jan. 27, 1922, p. 97, but the above is sufficiently accurate in most cases.

An empirical formula for this clearance (δ) in reaction turbines with radial clearances is

$$\delta = \frac{(15 + 0{\cdot}8d + 4h)}{1000} \text{ in.}$$

As δ is always small, it is sufficient in this formula to take approximate values of d and h.

Let
$$\left(h + \delta\frac{1-\beta}{\beta}\right) = h'$$
then
$$a = \pi\beta dh'$$

Substituting the above value for a in the equation for pressures—

$$p_1^2 - p_2^2 = \frac{NQ^2H}{209 \times 10^6 \times (\pi\beta dh')^2}$$
$$= \frac{1}{206 \times 10^7} \cdot \frac{Q^2H}{\beta^2} \cdot \frac{N}{h'^2d^2}$$

or
$$\frac{N}{h'^2d^2} = 206 \times 10^7\frac{\beta^2}{Q^2}H(p_1^2 - p_2^2)$$

If η is the (assumed) overall efficiency of the turbine and alternator, and I the total adiabatic heat-drop through the steam turbine (from p. 97)

$$Q' = \frac{3412}{\eta I} \times \text{K.W.}$$

or allowing x per cent. for dummy leakage, etc., the steam passing through the blades

$$Q = \frac{3412}{\eta I} \text{ K.W.} \left(\frac{100 - x}{100}\right)$$

In drawing up a table the method of procedure is similar at the start to that of the impulse turbine, except that the number of sections (n) is chosen arbitrarily as from practical experience. Values of H and H_0 are calculated as before, and intermediate values for each section obtained as in Fig. 75.

Since the constant $C = \dfrac{778\Delta I}{2{\cdot}3}$ (see p. 145), an approximate value can be determined by evaluating $\Delta I = \dfrac{I}{n} +$ say 3 per cent. for reheat.

Log X can then be adjusted to make $\Sigma \log X = \log X_1$, or $\Sigma\dfrac{1}{H}$; alternatively $\Sigma\dfrac{1}{H}$ can be evaluated.

Values for $\dfrac{N}{h^2d^2}$ are then calculated from the resulting pressures and steam consumption on the assumption that the blade opening $\beta = 0{\cdot}35$.

On p. 135 it was shown that the velocity ratio for each section of a reaction turbine

$$a = \frac{u}{c} = \sqrt{\frac{\Delta K}{1{\cdot}32\Delta I}}$$

where K = the turbine constant

$$= NR^2d^2 \times 10^{-9} \quad (R = \text{R.P.M.})$$

This velocity ratio may be estimated from Fig. 73, or the table on p. 133, when

$$\Delta K = 1{\cdot}32\Delta I a^2$$

and the total turbine constant K

$$K = n\Delta K$$

In each section

$$Nd^2 = \frac{\Delta K}{R^2} \times 10^9 = \text{a constant}$$

Now assume spindle diameters and choose by trial and error mean blade-ring diameters (d), such that values of N, d, and h' approximate as close as possible to the two relations

$$Nd^2 = \text{a constant}$$

and

$$\frac{N}{h'^2d^2} = 206 \times 10^7\frac{\beta^2}{Q^2}H(p_1{}^2 - p_2{}^2)$$

The values thus obtained should be rounded off so that N is a multiple of $\frac{1}{2}$, and the blade heights are even $\frac{1}{8}$ in. in the small sizes and even $\frac{1}{4}$ in. in the longer blades. If, as often happens, the last blade heights become much more than $\frac{d}{5}$, then "semi-wing" blades with $\beta = 0{\cdot}52$ and "wing" blades with $\beta = 0{\cdot}7$ should be substituted.

Provisional determination of the Proportions of a Parsons Turbine by the Homogeneous Head Method.—The following example of this method which was published in the aforesaid article in *Engineering*, may help to make this method clearer.

Type, reaction turbine with three cylinders.

Output, 3000 K.W.

Speed, 3000 R.P.M.

Steam conditions, 164·8 lb. per sq. in. absolute saturated, to 1 lb. or 28 in. vacuum (barometer 30 in.).

Using Peabody's Steam Tables (8th edit.)—

$$164{\cdot}8 \text{ lb. (abs.) } \phi = 1{\cdot}56 \text{ B.Th.U.} = 1193{\cdot}3 \quad V = 2{\cdot}75$$
$$1 \text{ lb. (abs.) } \phi = 1{\cdot}56 \text{ B.Th.U.} = 871{\cdot}1$$

adiabatic heat drop 322·2 B.Th.U. per lb.

Assume an internal efficiency of 75 per cent., the B.Th.U. in exhaust will equal

$$1193{\cdot}3 - 0{\cdot}75 \times 322{\cdot}2 = 951{\cdot}7$$

which corresponds to $\phi = 1{\cdot}703$ and $V_0 = 283$

$$H = 144 \times 164{\cdot}8 \times 2{\cdot}75 = 65{,}400$$
$$H_0 = 144 \times 1 \times 283 = 40{,}752$$

If the turbine is divided into 12 sections the adiabatic heat drop in each will be $\frac{322 \cdot 2}{12} = 26 \cdot 85$, or allowing 3 per cent for reheat $= 27 \cdot 66$. This gives an approximate value for C

$$C = \frac{778 \times 27 \cdot 66}{2 \cdot 3} = 9349$$

If an overall efficiency of 70 per cent. is assumed

$$Q' = \frac{3412}{322 \cdot 2 \times 0 \cdot 7} \times 3000 = 45{,}600 \text{ lb. per hour}$$

Allowing 5 per cent. for leakage, etc.

$$Q = 45{,}600\left(\frac{100 - 5}{100}\right) = 43{,}320$$

Hence, taking $\beta = 0 \cdot 35$

$$\frac{N}{h'^2 d^2} = \frac{206 \times 10^7 \times 0 \cdot 35^2}{43{,}320^2} \cdot \frac{p_1^2 - p_2^2}{H}$$
$$= 0 \cdot 1345 \frac{p_1^2 - p_2^2}{H}$$

The following table shows the result of working out this example, intermediate values of H being obtained by the method shown in Fig. 75.

TABLE VI.

Stage.	H.	Log X C = 9349.	Log X.	Log p.	p.	$\frac{N}{h'^2 d^2}$
	65,400			2·2170	164·8	
1	63,346	0·1476	0·1492	2·0678	116·9	0·0286
2	61,292	0·1525	0·1542	1·9136	81·9	0·0153
3	59,238	0·1578	0·1594	1·7542	56·8	0·0079
4	57,184	0·1635	0·1652	1·5890	38·8	0·00405
5	55,130	0·1696	0·1712	1·4178	26·2	0·00200
6	53,076	0·1767	0·1784	1·2394	17·4	0·00097
7	51,022	0·1832	0·1848	1·0546	11·3	0·00046
8	48,968	0·1908	0·1925	0·8621	7·28	0·000204
9	46,914	0·1993	0·2009	0·6612	4·59	0·000092
10	44,860	0·2084	0·2101	0·4511	2·83	0·000039
11	42,806	0·2184	0·2200	0·2311	1·70	0·0000161
12	40,752	0·2294	0·2310	0·0000	1·00	0·0000063
	Σ log X	2·1972	2·2169			
	log X_1	2·2169				
	12)	0·0197				
		0·00165				

In this table, with the approximate value of C = 9349, Σ log X comes out to 2·1972 instead of $\log \frac{p_1}{p_0} = \log \frac{164\cdot8}{1} = 2\cdot2169$. One-twelfth of the difference, therefore, is added to each approximate value of log X to get the true value, so that $\Sigma \log X = \log \frac{p_1}{p_0} = 2\cdot2169$.

If a velocity ratio (α) of 0·75 is assumed, the turbine constant for each section

$$\Delta K = \frac{1\cdot32 \times \Delta I}{\alpha^2}$$

$$= \frac{1\cdot32 \times 27\cdot66}{0\cdot75^2} = 20\cdot6$$

The value 1·32 is taken for the reaction turbine instead of 2·65 as in the impulse turbine, since there are 2N rows reckoned in both rotor and casing. The total K for the turbine is 12 × 20·6 = 247.

For each section

$$Nd^2 = \frac{\Delta K}{R^2} \times 10^9$$

$$= \frac{20\cdot6 \times 10^9}{3000^2} = 2280$$

The following table gives the blading of such a turbine, the values of $\frac{N}{h'd^2}$ being repeated from the table before :—

TABLE VII.

Stage.	$\frac{N}{h'd^2}$	Spindle diam.	Assumed mean diam.	N.	h'.	Proposed N	Proposed h
1	$2\cdot86 \times 10^{-2}$	14	15·2	9·9	1·22	10	$1\frac{1}{8}$
2	$1\cdot53 \times 10^{-2}$	14	15·6	9·4	1·59	9	$1\frac{1}{2}$
3	$79\cdot0 \times 10^{-4}$	14	16·1	8·8	2·07	9	2
4	$40\cdot5 \times 10^{-4}$	20	21·6	4·9	1·61	5	$1\frac{1}{2}$
5	$20\cdot0 \times 10^{-4}$	20	22·1	4·7	2·19	5	$2\frac{1}{8}$
6	$970\cdot0 \times 10^{-6}$	20	23·0	4·3	2·90	4	$2\frac{7}{8}$
7	$460\cdot0 \times 10^{-6}$	29	31·3	2·33	2·28	$2\frac{1}{2}$	$2\frac{1}{4}$
8	$204\cdot0 \times 10^{-6}$	29	32·2	2·20	3·24	2	$3\frac{1}{4}$
9	$92\cdot0 \times 10^{-6}$	29	33·5	2·04	4·46	2	$4\frac{1}{2}$
10	$39\cdot0 \times 10^{-6}$	29	35·2	1·84	6·20	2	$6\frac{1}{2}$
11	$16\cdot1 \times 10^{-6}$	29	35·2	1·84	9·62	2	$6\frac{1}{2}$X
12	$6\cdot3 \times 10^{-6}$	29	35·2	1·84	15·4	$1\frac{1}{2}$	$6\frac{1}{2}$XX

The spindle diameters (s) are assumed as H.P. = 14 in., I.P. = 20 in., and L.P. = 29 in., or roughly in the ratio 1 : $\sqrt{2}$: 2. The total K is divided up to give $\frac{1}{4}$K in the H.P. cylinder, $\frac{1}{4}$K in the I.P. cylinder, and $\frac{1}{2}$K in L.P. cylinder. The mean diameters of the blade rings (d)

$= s + h'$. If this is estimated for the first section to be 15 in., then

$$N = \frac{2280}{15^2} = 10{\cdot}14$$

and
$$\frac{10{\cdot}14}{h'^2 \times 15^2} = \frac{0{\cdot}1345(164{\cdot}8^2 - 116{\cdot}9^2)}{63{,}346} = 0{\cdot}0286$$

whence $h' = 1{\cdot}26$ ins. or $d = 14 + 1{\cdot}26 = 15{\cdot}26$ in.

A second attempt taking $d = 15{\cdot}2$ in. gives $N = 9{\cdot}9$ and $h' = 1{\cdot}22$ in. and this one has been tabulated. A repetition of this method gives the corresponding values for the other sections. The last row but one (marked X) will have to be a semi-wing blade in which the opening $\beta = 0{\cdot}52$, and the last row (marked XX) a wing-blade in which $\beta = 0{\cdot}70$.

It will be noted that this method, like all others for reaction turbines, involves a certain amount of adjustment, but this is unavoidable, as a complicated cubic equation has to be solved.

CHAPTER VIII

SOME PROBLEMS ARISING OUT OF STEAM TURBINE DESIGN

General Summary.—The development of the design of the steam turbine is a beautiful example from the engineering world of the direct application of theory to practice. The turbine is at once the simplest and the most complicated of our heat engines; simple, because there is no intermediary between the working fluid and the rotary motion from which it delivers its power; complicated, because that working fluid is steam.

It has inspired, and in fact required, a considerable amount of research, both of an analytical and experimental nature, to enable it to reach the very prominent position it holds to-day. For instance, early designs and calculations showed that high rotational speeds were conducive to high efficiencies, besides reducing the size and therefore the cost of a machine for any required output. This necessitated investigation into the design of dynamos and alternators to run at much higher speeds than had previously been considered possible. Again, owing to leakage past the tips of the blades in the reaction type, or to the increase of friction resistance (due to the increased velocity of the steam) in the impulse type, there is little to be gained, except in large sizes, in using initial steam pressures above 250 lb. per sq. in. (abs.). But *superheated steam* leads to considerable economy. Initial condensation is almost entirely absent in turbine work, but the fluid friction is much less with superheated or dry steam than it is with steam containing moisture. This has led to the increased importance of ascertaining the specific heats of superheated steam, which, as pointed out in the chapter on boilers (p. 4), are not constant. The experiments of Knoblauch and Jakob,[1] in Munich, were long looked upon as the most important investigation in this matter, but, as so often happens with German work, their deductions, based on extrapolation, were anything but consistent, and the most authentic results so far obtained are due to Professor H. L. Callendar, whose work [2] is based on a characteristic equation which is thermodynamically correct and in which the constants were determined from careful experiments.

Again, on account of fluid friction and the size of the L.P. cylinder which would be required, expansion in a steam engine is rarely carried

[1] *V. D. I.*, vol. li. (1907), pp. 81 and 124.

[2] "Properties of Steam" (1921), by H. L. Callendar, F.R.S., chap. vi. See also Ewing's "Thermodynamics for Engineers" (1920), p. 318.

beyond 2 lb. per sq. in. (abs.). But a turbine can extract work at a much lower pressure, and so it follows that a high vacuum is conducive to economy. This has emphasised the importance of *condenser design* and led to such devices as the use of the steam-jet augmenter, which consists of a pipe leading from the bottom of the main condenser into a small condenser, with a steam ejector in the pipe, which extracts nearly all the residual air and vapour from the main condenser.

This capacity for recovering a comparatively large amount of energy from the lower pressures of steam has led to the introduction of exhaust-pressure turbines, which utilise steam after it has passed through a reciprocating engine. Such turbines also lend themselves to cases where steam is used intermittently. Here their use has been rendered possible by regenerative steam accumulators, with which the name of Professor Rateau will always be associated.

Mechanically the turbine has brought the question of the high-speed bearing and its lubrication into prominence, as well as the design of reduction gearing for large powers. The first has led to the design of the Michell[1] pivoted bearing, which automatically adjusts itself to the oil film; the second to the De Laval, Parsons, and McAlpine helical gearing,[2] which have enormously increased the utility of the turbine.

Problems in design have led to the importance of understanding the whirling speed of shafts, and here the work of the late Professor Dunkerley,[3] and of K. Baumann,[4] Dr. Arthur Morley [5] and others, has proved invaluable. Stresses set up in turbine blading have been treated by Professor Gerald Stoney,[6] and the strength of rotating discs can now be obtained from curves prepared by W. Knight,[7] which are based on the fundamental equations of Professor Stodola. The vibrating of rotating discs and of the blading has also to be taken into consideration, but so far this problem does not appear to have been properly investigated.

Lastly, the use of high superheats has led to collaboration with metallurgists on the phenomenon known as the growth of cast iron under high temperatures. In this connection experience shows that it is unwise to subject cast iron to steam temperatures over 500° F. This necessitates steel castings for the high-pressure ends of some large turbines.

Condensers.—Although it is outside the scope of such a book as this to describe in detail the numerous accessories of heat engines and boilers, one or two exceptions will be made in cases where published information is either scanty or difficult of access to the general engineer. This applies particularly to the theory and design of the modern surface

[1] See "The Problem of the Thrust Bearing," by H. T. Newbigin, *Proc. Inst. C.E.* vol. cxcvi. (1914), p. 223; or "Steam Turbines," by W. J. Goudie (1917), p. 319.

[2] See "High-speed Turbine Gears," by Gerald Stoney, F.R.S., *Proc. Manchester Association of Engineers* (1919–1920), reprinted in *Engineering*, vol. cviii. (1919), p. 729.

[3] *Philosophical Trans. of the Royal Society* (1894), vol. clxxxv. A.

[4] *Journal Inst. E.E.* vol. xlviii. (1912), p. 807.

[5] *Engineering*, vol. lxxxviii. (1909), pp. 135 and 205.

[6] *Engineering*, vol. cv. (1918), p. 447.

[7] *Engineering*, vol. civ. (1917), p. 109.

condenser, which most text-books treat with meagre respect, and which as yet cannot be said to have any literature of its own.

The main object of the condenser is to produce and maintain a vacuum sufficiently high to get the best efficiency out of the steam engine or turbine for which it is designed. The surface condenser, in addition, returns the condensed steam or condensate back to the boiler feed without intermingling it with the cooling water, as would be the case with the jet type of condenser.

The height of the vacuum possible in any installation is controlled by the quantity and the temperature of cooling water available. Where river cooling or the sea can be used it pays to have the highest vacuum that the plant will give. With a modern steam turbine the gain in economy may be estimated roughly from the following table :—

Vacuum. Hg in.	Corresponding temp. ° F.		Per cent. improvement in consumption.	° F. for 1 per cent. improvement.
26	127			
		11	4	2·7
27	116			
		14	6	2·3
28	102			
		22	9	2·4
29	80			

From which it can be seen that there is approximately 1 per cent. gain in economy for every extra 2·5° F.

In considering the design of a condenser the pressures and temperatures at the various points are to a large extent interchangeable. The one can be obtained from the other from steam tables. In the numerical examples cited here Peabody's Tables (8th ed., 1914) have been used.

The objects of a good design are to keep the drop in pressure (or in temperature) through the condenser as small as possible, and at the same time to make the difference in temperature between the incoming steam and the inlet cooling water a minimum.

If T is the temperature in ° F. corresponding to the vacuum and t_c the temperature of the condensate, then

$$t_c = \mathrm{T} - 10^\circ \text{ F. in a good condenser}$$

Every ten degrees more means approximately 1 per cent. more coal in the boilers.

Again, if t_1 is the incoming temperature of the cooling water

$$\mathrm{T} - t_1 = 22^\circ \text{ to } 25^\circ \text{ F.}$$

in a good design, though many condensers working at present have $\mathrm{T} - t_1$ as high as 40° or 45° F.

Factors in a good Condenser.—

Let
$$\mathrm{X} = \frac{\text{water}}{\text{steam}} = \frac{\mathrm{Q}'}{\mathrm{Q}}$$

where Q′ = quantity of cooling water in lb. per hour
and Q = quantity of steam condensed in lb. per hour
also let S = the lb. of steam condensed per sq. ft. of surface per hour

$$= \frac{\mathrm{Q}}{\mathrm{A}}, \text{ where A = external area of tube surface in sq. ft.}$$

Let K = the conductivity of the tubes in B.Th.U. per sq. ft. of cooling surface per 1° F. per hour

then $Q = SA$, and $Q' = SAX$

Consider an element of a condenser tube—

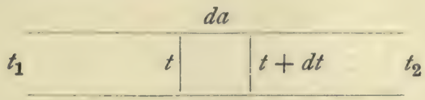


FIG. 76.

$$d(\text{B.Th.U.}) = SAXdt = K(T - t)da$$

where T = temperature of steam corresponding to the vacuum (from tables)

and t = temperature of cooling water at any point along the tube

$$Kda = SAX\frac{dt}{T - t}$$

$$KA = SAX\int_{t_1}^{t_2}\frac{dt}{T - t}$$

or

$$K = SX\int_{t_1}^{t_2}\frac{dt}{T - t}$$

$$K = SX \log_e \frac{T - t_1}{T - t_2}$$

$$= 2{\cdot}3\ SX \log \frac{T - t_1}{T - t_2}$$

which gives a formula for the conductivity of a surface condenser.

In a good modern design

X = 40 to 60
S = 8 to 12
and K = 600 to 800, for a good vacuum.

With a bad vacuum or a dirty condenser the conductivity (K) may be as low as 150 – 200.

It may be noticed that approximately 1000 B.Th.U. are required to condense 1 lb. of steam,

so that
$$t_2 - t_1 = \frac{1000}{X}$$

EXAMPLE :—

Let X = 60, and the temperature of the inlet cooling water

$$t_1 = 60° \text{ F.}$$

then the temperature of the outlet cooling water

$$t_2 = \frac{1000}{60} + t_1 = 17 + 60 = 77° \text{ F.}$$

(neglecting fractions of a degree)

Take S = 10 lb. of steam per sq. ft. of cooling surface per hour, and assume a vacuum of 28·8 in., which corresponds to 0·6 lb. per sq. in., or 85° F. (= T), then

$$K = 2·3 \times 10 \times 60 \times \log \frac{85 - 60}{85 - 77}$$
$$= 680$$

and
$$T - t_1 = 85 - 60 = 25° \text{ F.}$$
$$T - t_2 = 85 - 77 = 8° \text{ F.}$$

which is a satisfactory result.

Supposing that the conductivity (K) falls to 200 due to a dirty condenser or to air leaks, then

$$200 = 2·3 \times 60 \times 10 \times \log \frac{T - t_1}{T - t_2}$$

or
$$\log \frac{T - t_1}{T - t_2} = \frac{200}{1380} = 0·145$$

so that
$$\frac{T - t_1}{T - t_2} = 1·4$$

then if the temperature t_1 and t_2 of the cooling water remain the same

$$T - 60 = 1·4\ (T - 77)$$

and
$$T = 120°, \text{ or } 1·7 \text{ lb. per sq. in}$$

which means a vacuum of only 26·6 in. instead of 28·8 in.

Air in the Condensate.—The problem of condenser design is complicated by the fact that air is always present in the condensate. The bulk of this air enters the system through leakage. In a good turbine plant it may be taken as about 37 lb. of air in every 100,000 lb. of steam. With steam engines this proportion may be 70 lb. of air per 100,000 lb. of steam. In badly designed or in old plants such figures are much exceeded.

The effect of this air will now be considered.

Let p_c be the pressure in lb. per sq. in. of the vapour at the bottom of the condenser.

By Dalton's law this pressure is the sum of the partial pressures of the air (p_a) and of the steam vapour (p_s)

or
$$p_c = p_a + p_s$$

p_s corresponds to the temperature (t_s) of the vapour drawn off from the condenser in the air pump.

If δ is the drop in pressure through the condenser (which should not exceed $\frac{1}{4}$ in. to $\frac{3}{8}$ in. of mercury), and p_0 is the exhaust pressure of the turbine, which corresponds to T, then

$$p_o - \delta = p_a + p_s = p_c$$

Again, let d = lb. of air per lb. of steam

V_1 = volumes in cub. ft. per lb. of air at 1 lb. per sq. in. pressure, and 90° F. temperature t_s (206 cub. ft. per lb.)

V = the volume of air which by Dalton's law = the volume of steam vapour

then
$$V = \frac{V_1}{p_a} d \times \frac{t_s + 460}{90 + 460}$$

which gives an equation for calculating the volume of the air to be extracted in cub. ft. per lb. of steam, and enables the capacity of the air pump to be estimated.

To take an example—

Assume a vacuum of 28·8 in. so that $p_0 = 0{\cdot}6$ lb. per sq. in. Let the drop in pressure (δ) through the condenser be 0·2 in., then $p_c = 28{\cdot}8 + 0{\cdot}2 = 29$ in., or 0·5 lb. per sq. in.

Suppose the air vapour cools down 5° F. below the temperature of the condensate (t_c) corresponding to p_c, namely, 80° F., so that $t_s = 75°$ F., then from the tables, $p_s = 0{\cdot}43$ lb. per sq. in., and the pressure of the air $p_a = p_c - p_s = 0{\cdot}5 - 0{\cdot}43 = 0{\cdot}07$ lb. per sq. in.

If d is taken at 37×10^{-5}, then the volume of air to be extracted per lb. of steam (V)

$$= \frac{206}{0{\cdot}07} \times 37 \times 10^{-5} \times \frac{535}{550} = 1{\cdot}06 \text{ cub. ft.}$$

Suppose now the vapour is reduced in temperature by means of a cooler to 70° F., then $t_s = 70°$ F., and $p_s = 0{\cdot}36$ lb. per sq. in., the pressure of the air $p_a = 0{\cdot}5 - 0{\cdot}36 = 0{\cdot}14$ lb. per sq. in.

and
$$V = \frac{206}{0{\cdot}14} \times 37 \times 10^{-5} \times \frac{530}{550} = 0{\cdot}525 \text{ cub. ft.}$$

and the capacity of the air pump can be halved, or alternatively the same air pump can deal with twice the quantity of condensate.

The Council of the North-East Coast Institution of Engineers and Shipbuilders appointed a sub-committee consisting of Messrs. E. L. Orde, C. W. Cairns, and J. Morrow, to inquire into various methods of producing high vacuum by experimental research in an apparatus fitted up by Messrs. Richardsons, Westgarth & Co., of Hartlepool, and their first report,[1] presented during the session of 1916–17, contains valuable information on various methods of the extraction of air from condensers. The apparatus was arranged so that the quantity of air present in the condenser could be regulated, and progressive experiments were carried out when the air was being withdrawn from the condenser by means of reciprocating air pumps. These pumps could be connected up on either the "all-wet" system or on the "wet-and-dry" system with an air cooler.

The results of their tests are shown diagrammatically in Figs. 77 and 78, reproduced here by the courtesy of the Council of the N.E.C. Inst. The curves marked A, B, and C in Fig. 77 embody the tests on the all-wet system with two pumps, where the total condensate was 13,600, 43,000, and 58,000 lb. per hour respectively. It should be noted that these results lie approximately on three parallel straight lines, the smallest quantity of condensate giving the best vacuum. Curve D shows a wet system with a single barrel, using 31,000 lb. of condensate per hour.

[1] Published by the Institution, Bolbee Hall, Newcastle-upon-Tyne.

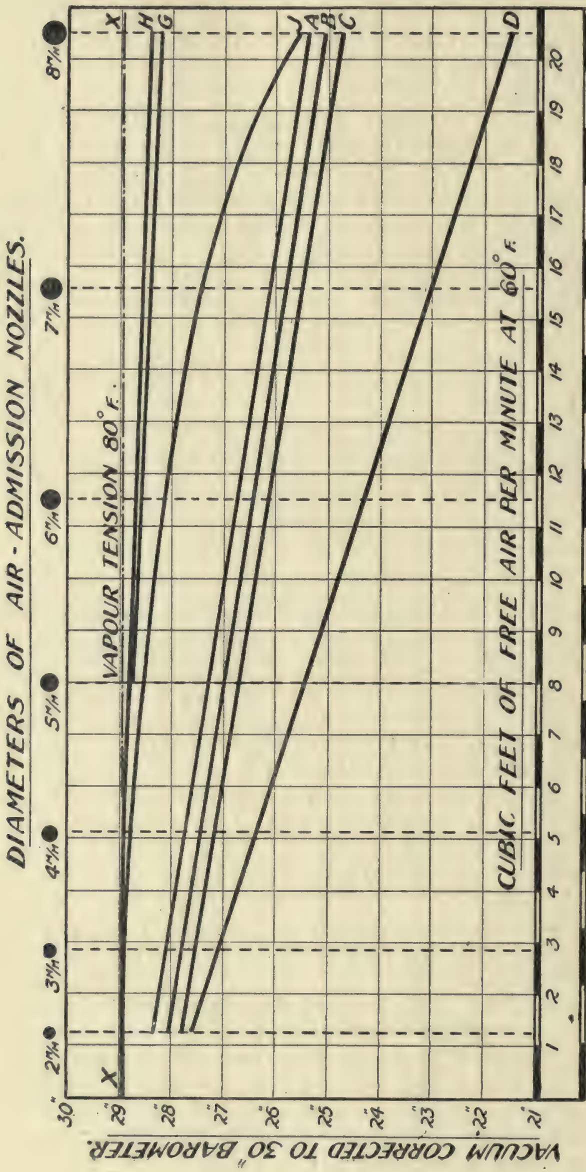


FIG. 77.—N.E.C. Inst. condenser tests. "All-wet" system.

Curves E and F on Fig. 78 show the wet-and-dry system, for 58,500 and 55,600 lb. per hour condensate respectively. In E the inlet temperature of the circulating water to the air cooler was 50° F., and in F 70° F.

The ideal vacuum of 28·97 in. is shown in both figures by the vapour tension curve marked X. This is derived from the temperature of the condensate t_c (80° F.), and represents the vacuum which would be obtained if no air were present. During the tests t_c was kept constant.

It should be noted that air pumps alone are only able to maintain a vacuum of from $\frac{1}{2}$ in. to 1 in. lower than that which might be theoretically expected of them.

Kinetic Steam-jet Augmenter.—The exhaustion of air and steam vapour can be augmented by means of a steam-jet ejector placed close to the condenser at the point where the air and vapour is extracted, the vapour and steam being condensed and cooled by an auxiliary condenser before entering the air-pump. This will have the effect of increasing the pressure (p_c), and the result can best be illustrated by an example.

Supposing the steam jet increases p_c from 0·5 lb. per sq. in. to 1·5 lb. per sq. in. (*i.e.* three compressions). This is equivalent to altering the vacuum to be dealt with by the air pump from 29 in. to 27 in.

Assume that the temperature of the air and vapour leaving the auxiliary condenser is 82° F. (t_s). This corresponds to 0·54 lb. per sq. in. (p_s), so that the air pressure $p_a = 1{\cdot}50 - 0{\cdot}54 = 0{\cdot}96$ lb. per sq. in. Then

$$V = \frac{206}{0{\cdot}96} \times 37 \times 10^{-5} \times \frac{542}{550} = 0{\cdot}078 \text{ cub. ft.}$$

as against 1·06 cub. ft. in the first example, so that a very much smaller air pump will suffice.

In the report on certain methods of producing a vacuum referred to above, experiments were carried out with this kinetic-jet system, in which air was withdrawn from the condenser by means of a steam jet or jets, by which it was compressed and delivered to the air pumps in a less rarefied state.

In the apparatus used this air was discharged by the jets into a second and smaller receiver or jet condenser, in which the jet steam was condensed by a part of the condensate and the heat recovered for the feed. Some of the results are shown in Fig. 77, in curves H, G, and J. These represent the use of a kinetic jet working in conjunction with an all-wet system. Curve H shows approximately 10,000 lb. of condensate per hour, curve G 60,000 lb. when working with a twin-barrel air pump, and curve J 25,000 lb. with a single-barrel air pump.

Curve K, in Fig. 78, shows the corresponding results with a kinetic wet-and-dry system working with 54,300 lb. of condensate per hour.

The improvement in every case is at once apparent, more especially as care was taken to carry out the two sets of tests under as similar conditions as possible. Unlike the air pumps above, nearly the ideal

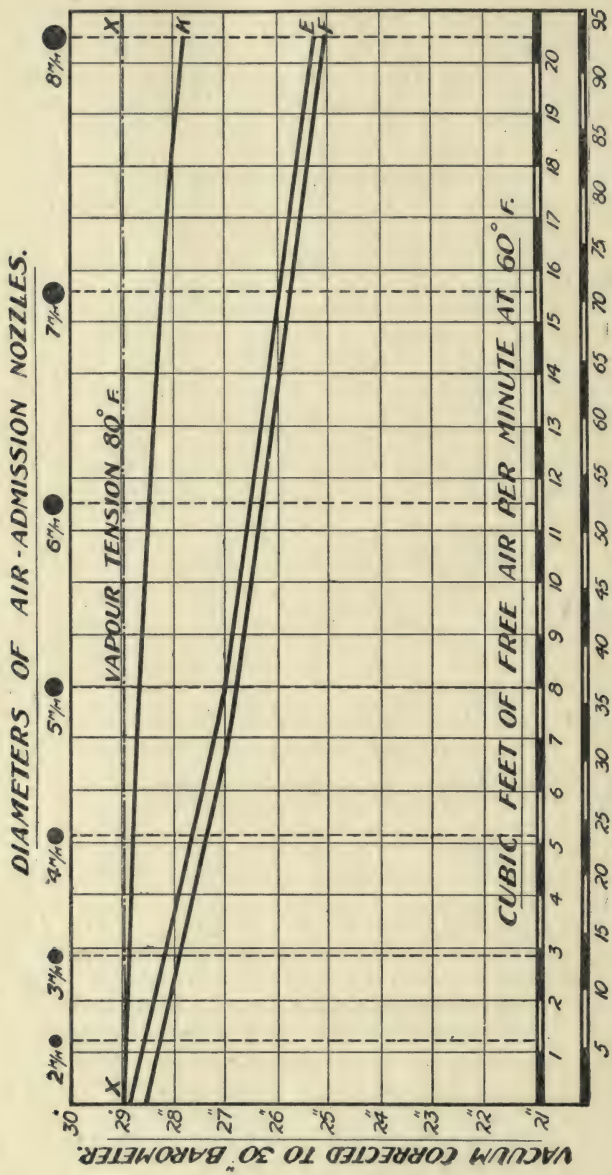


FIG. 78.—N.E.C. Inst. condenser tests. "Wet-and-dry" system.

vacuum is attained with small air leakages, etc., and there is a considerable amount of reserve capacity to deal with large leakages of air.

Power of Circulating Water Pumps.—In order to estimate the amount of power required by the circulating water pumps it is necessary to calculate the head due to friction through the condenser. This may conveniently be done as follows:—

Let d = inside diameter of tubes in in.

d' = outside diameter of tubes in in.

$$\frac{d'}{d} = \frac{6}{5} \text{ approximately in modern condenser tubes}$$

Q' = quantity of cooling water circulating in lb. per hour through one tube

v = velocity of flow in ft. per sec.

then the area of the tube

$$= \frac{\pi d^2}{4 \times 144} = \frac{d^2}{182} \text{ sq. ft.}$$

and

$$Q' = v \times 3600 \times \frac{d^2}{182} \times 62{\cdot}5 \text{ lb. per hour}$$

$$= 1240vd^2 \text{ lb. per hour} \quad \ldots\ldots\ldots \quad (1)$$

Again, if S = lb. of steam condensed per sq. ft. of cooling surface per hour

L = total length of flow in ft.

$$X = \text{ratio of } \frac{\text{cooling water}}{\text{steam condensed}} = \left(\frac{Q'}{Q}\right)$$

then

$$Q' = SX\frac{L\pi d}{12} \times \frac{d'}{d}$$

$$= SX\frac{L\pi d}{12} \times \frac{6}{5} = \frac{SLdX}{3{\cdot}15} \quad \ldots\ldots \quad (2)$$

Equating (1) and (2)

$$\frac{SLXd}{3{\cdot}15} = 1240vd^2$$

from which

$$d = \frac{SLX}{3900v} \text{ in.} \quad \ldots\ldots \quad (3)$$

From the researches of T. E. Stanton and J. R. Pannell,[1] of the National Physical Laboratory, who experimented on the flow of air, oil, and water through small brass pipes, it appears that approximately a head corresponding to the velocity of flow is lost in every 32 diameters of a pipe. This is sometimes known as the 32-diameter rule. For instance, since

$$H = \frac{v^2}{2g}$$

where H is the head in feet, then if v = 8 ft./sec.

$$H = \frac{8^2}{2 \times 32} = 1 \text{ ft. loss}$$

[1] Stanton and Pannell, *Phil. Trans.* A. vol. ccxiv. (1914).

so that the loss of head due to a velocity of 8 ft. per sec. in a 1-in. bore pipe 32 in. long may be taken as 1 ft.

Therefore in any condenser the loss of head due to this cause

$$H = \frac{v^2}{2g} \times \frac{L \times 12}{32d} = \frac{v^2 L}{d} \times \frac{1}{170} \quad . \quad . \quad . \quad . \quad (4)$$

substituting the value of d found in equation (3)

$$H = \frac{v^2 L}{170} \times \frac{3900v}{SXL} = 22 \cdot 6 \frac{v^3}{SX} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

which is independent of the size of the tubes, *i.e.* of d and L.

Alternatively from (3)

$$v = \frac{SLX}{3900d}$$

substituting this in equation (4) gives

$$H = \frac{S^2 L^2 X^2}{3900^2 d^2} \cdot \frac{L}{d} \times \frac{1}{170} = \frac{S^2 X^2 L^3}{d^3} \times \frac{1}{26 \times 10^8} \cdot \quad . \quad (6)$$

which is independent of the velocity of flow. It is found in practice that for each pass in the condenser one head due to the velocity should be added.

EXAMPLE :—

Let S = 10 lb. per sq. ft. per hour

$X = 60 \left(\text{ratio } \frac{\text{water}}{\text{steam}} \text{ or } \frac{Q'}{Q}\right)$

H = 10 ft.

$d = \frac{5}{8}$-in. bore ($\frac{3}{4}$-in. outside diameter)

then from (6) $$L^3 = \frac{10 \times (\frac{5}{8})^3 \times 26 \times 10^8}{60^2 \times 10^2} = 25^3 \text{ ft.}$$

making a double-flow condenser 12 ft. 6 in. long.

From (5) $$v^3 = \frac{10 \times 10 \times 60}{22 \cdot 6} = (6 \cdot 4)^3$$

so that the loss of head due to two bends

$$= 2 \times \frac{6 \cdot 4^2}{2g} = 1 \cdot 28 \text{ ft.}$$

and the total head in feet against which the circulating pump will have to act = 10 + 1·28 = 11·28 ft.

It is customary to limit the total head in condenser work to a maximum of about 14 ft. A head of 10 ft. is very common in modern practice.

Another point which has to be watched in the condenser design is the provision of a sufficiently high water velocity to ensure turbulence, otherwise the cooling effect is not nearly so effective. Stanton and

Pannell, in their paper quoted above, show that for water turbulent flow is certain to take place at the following velocities :—

tubes $\frac{5}{8}$ in. outside diameter $v = 5$ ft. per sec. at 50° F.
$= 2\frac{1}{2}$ ft. per sec. at 110° F.
„ $\frac{3}{4}$ in. „ „ $v = 4$ ft. per sec. at 50° F.
$= 2$ ft. per sec. at 110° F.
„ 1 in. „ „ $v = 3{\cdot}6$ ft. per sec. at 50° F.
$= 1{\cdot}9$ ft. per sec. at 110° F.

If care is taken to exceed these velocities it is safe to assume that turbulence is present.

CHAPTER IX

SOME POINTS IN THE DESIGN OF THE RECIPROCATING STEAM ENGINE

THE theory and design of the reciprocating steam engine has engaged the attention of more master minds in the engineering world than any other kind of heat engine. The problem has been minutely investigated from every conceivable point of view, with the result that the modern designer is in the happy position of being able to turn for information in all directions. From the time of Rankine, theory and practice have worked hand in hand, not only on the behaviour of steam when constrained to work behind a piston in a cylinder, but also on the numerous mechanical problems which arise out of the movements of heavy masses of metal.

So much work has been published on the subject that there is a danger of the student or the young engineer obtaining an exaggerated idea of the importance of this type of heat engine as a prime mover. The steam engine still remains supreme where reliability, flexibility, and large starting torques are of prime importance. Such conditions are to be found in locomotives, marine propulsion, and special types of stationary engines, such as are used for colliery winding, rolling mills, or works requiring auxiliary steam. Where economy in operation is a predominating factor the reciprocating steam engine no longer holds the field, but is being gradually replaced by other forms of heat engine.

For reasons such as these it is hardly necessary to do more than indicate here the general trend of modern practice in designing stationary steam engines, and to point out at the same time some of the salient difficulties that have been overcome in the past.

The Trend of Design.—The different forms that a steam engine may take have been briefly referred to on p. 103.

Considering under this heading the various ways in which economy may be effected, it may be pointed out that in the past this has been obtained by the use of (*a*) condensers; (*b*) higher boiler pressures and the use of superheated steam; (*c*) jacketing the cylinders with steam; (*d*) compounding or expanding the steam in more than one cylinder, and (*e*) improvements in mechanical design. These methods may conveniently be considered separately.

(*a*) *Condensers.*—A glance at the Mollier diagram for steam, referred to on p. 96, shows the advantage to be gained in expanding steam below the pressure of the atmosphere by the use of a condenser. Lay

a scale vertically on the diagram to pass through any given point on the condition curve, corresponding to the initial state of the steam, and note how the lines of constant pressure get further and further apart as the pressure decreases. For instance, for an entropy (ϕ) of about 1·57 the difference between 150 lb. (abs.) and 15 lb. (abs.) is 171 B.Th.U., whereas if the steam is condensed to 2 lb. (abs.) the difference is 283 B.Th.U., or a gain of 112 B.Th.U. in the available heat. In the same way the gain in theoretical consumption can be seen on the curves shown in Fig. 53 on p. 98, which are derived from the Mollier chart.

In the steam engine there is, however, a practical limit to the size of the vacuum which it pays to install. In round figures atmospheric pressure corresponds to 15 lb. per sq. in., or to 30 in. of mercury,[1] so that roughly every 2 in. of vacuum represents one lb. decrease in steam pressure. But as the pressure drops the specific volume of the steam, that is, the number of cub. ft. occupied per lb. of steam, increases very rapidly. This can be seen from t-ϕ steam tables. At 15 lb. (abs.) $\phi = 1{\cdot}57$, one lb. of steam occupies approximately 23 cub. ft., but if the pressure is reduced to 2 lb. (abs.), or to a vacuum of $2(15 - 2) = 26$ in., this volume is increased adiabatically to about 136 cub. ft., which means that the capacity of the L.P. cylinder will have to be six times as large to contain the same quantity of steam. In the example just quoted, if expansion had taken place down to 1 lb. (abs.) the total B.Th.U. available would have been 321 as against 283 at 2 lb. (abs.); but the volume of the steam would be about 260 cub. ft. as compared with 136 cub. ft., and the capacity of the L.P. cylinder would have to be nearly doubled again. For this reason reciprocating steam engines are rarely designed to work with a higher vacuum than 27 in., and very frequently do not exceed 25 in.

The effect of varying the vacuum is clearly brought out in some tests which were carried out in 1913, jointly by Messrs. Belliss & Morcom, Ltd., of Birmingham, and by the University of Birmingham, on an engine built by the former for the latter. This engine was a small "V"-type compound (see Fig. 69, p. 127) of 70 K.W. capacity, designed to run at 500 R.P.M., and the following curves are reproduced by the courtesy of Messrs. Belliss & Morcom. The object of these tests was to find how the steam consumption improved as the vacuum was increased from atmospheric pressure to 26 in. under varying load or varying steam pressure. The comparative results are shown in Figs. 79 and 80. In both cases there is a steady improvement, which amounts approximately to a gain of one per cent. in the steam consumption for each additional inch of vacuum.

Except with small engines, or when steam is required for other purposes, condensing is almost universal in British and European practice. In America, on the other hand, non-condensing plants are much more common.

(*b*) *Higher Boiler Pressures and Superheated Steam.*—The curves in Fig. 53 on p. 98 show that little gain in theoretical consumption can

[1] More exactly, 30 in. of mercury corresponds to 14·690 lb. per sq. in. at the latitude of London and at a temperature of 62° F. or 17° C. (Callendar).

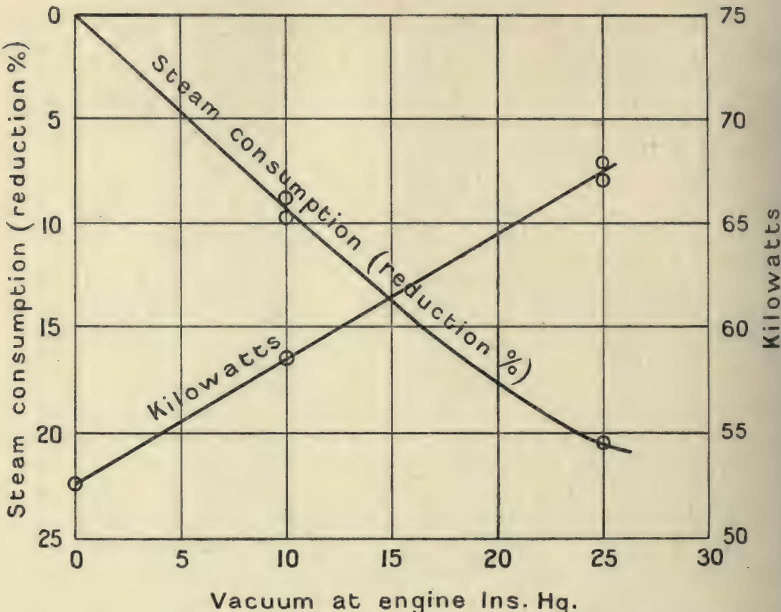


FIG. 79.—Steam-engine vacuum tests with varying load and constant pressure (177 lb. sq. in. (gauge) at the chest).

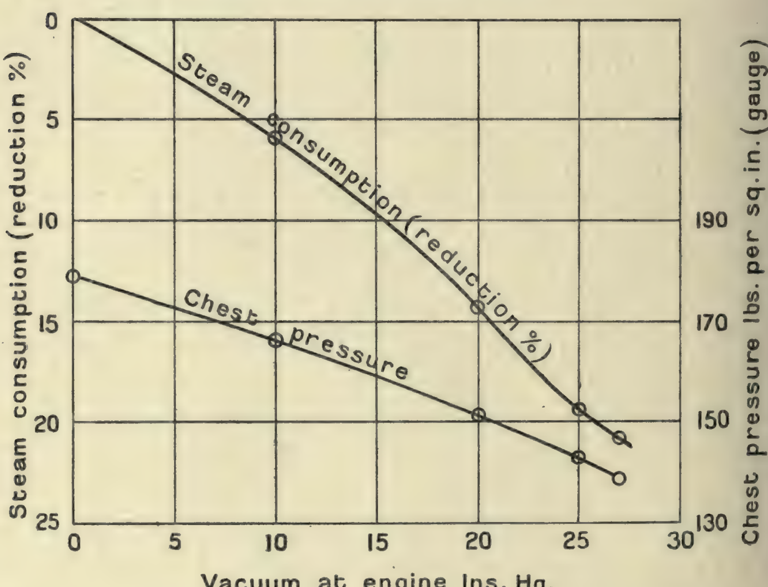


FIG. 80.—Steam-engine vacuum tests with varying pressure and constant load (52 K.W.).

be expected with boiler pressures higher than 200 lb. per sq. in. (abs.) in condensing engines, and this is borne out in practice. The improvement is more marked below 150 lb. per sq. in. (abs.), and early designers, who were working in this region, found it to their advantage to study mechanical details to enable their plants to work with higher pressures. Modern practice does not, as a rule, exceed 175 lbs. per sq. in. (abs.) for smaller compound engines, either condensing or non-condensing, and 220 lb. per sq. in. (abs.) for large triple-expansion engines.

Fig. 53 also shows that the gain in theoretical consumption by superheating is not nearly so great as by condensing. This can be seen on the Mollier diagram by taking the pressure point for the vertical scale on the desired line of superheat and comparing it with the corresponding point on the saturation curve.

But superheat has the effect of reducing *initial condensation* very considerably, and also of lessening what is sometimes called the *missing quantity*. Initial condensation depends on the amount of clearance, and continues up to the point of cut-off. It also depends on the mean temperature of the cylinder walls and the time of contact with them. As the piston goes forward this water is re-evaporated, but of course there is a reduction in the work done by the steam. After cut-off more steam is condensed as a result of expansion, but all this wetness is removed by the end of the exhaust stroke.

If a saturation curve be drawn on the indicator card to represent the expansion if the steam remains dry throughout the stroke, it will usually lie outside the actual expansion curve of the diagram. Such curves may be seen marked S and S′ in Fig. 84 on p. 178. By drawing horizontal lines across the diagram at any point between cut-off and release, the part intercepted by these two expansion curves forms a measure of the "missing quantity." This missing quantity in the past has been attributed to initial condensation, but the work of Professors Callendar and Nicolson,[1] and subsequent investigations published in the first report by Prof. Capper, of the Steam-Engine Research Committee,[2] point to leakage past the valves as being a factor which must also be taken into account. The whole question is one of the few interesting problems connected with steam engines which cannot as yet be considered as settled. The present position is fully discussed by Inchley, in his book on the "Theory of Heat Engines,"[3] where references are also made to a number of other papers bearing on the subject.

The improvement in consumption actually effected by superheat may be seen from Fig. 81, which is included here by kind permission of Messrs. Belliss & Morcom, Ltd., and represents the results of other tests carried out on the Birmingham University engine mentioned above. Even with such a comparatively small engine the consumption per K.W. is reduced from 29·5 lb. per K.W. hr. with no superheat to 20 lb. with 170° F. superheat, an improvement of 28 per cent. As these results are fairly typical of what may he expected with engines having common steam and exhaust ports, Fig. 82, which is derived

[1] *Proc. Inst. C.E.* vol. cxxxi. 1898.
[2] *Proc. Inst. M.E.* Parts i. and ii. 1905, p. 171.
[3] Second edit., 1920, pp. 112–129.

from Fig. 81, may be of use in showing the improvement per cent. for every 10° F. of superheat.

A consideration of the problem shows that superheat is most effective with designs which otherwise would show the largest missing

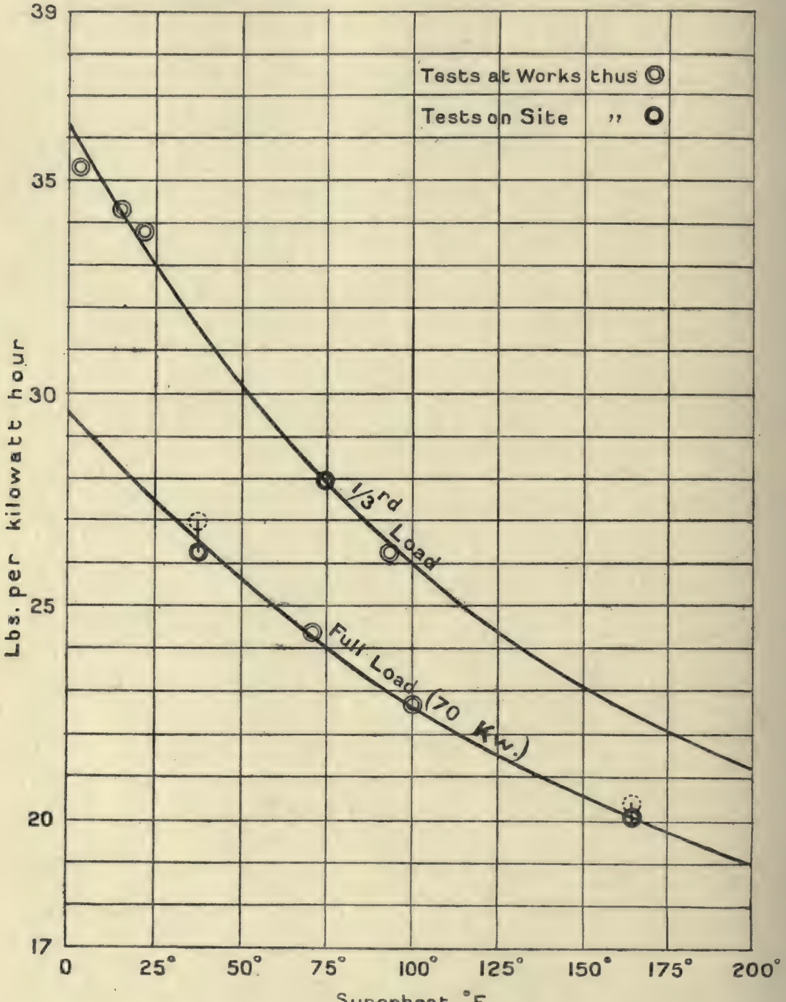


FIG. 81.—70 K.W. 500 R.P.M. steam-engine tests, showing affect of superheat. Steam conditions, 170 lb. sq. in. gauge, 26 in. vacuum.

quantity, that is to say, with engines having large clearance surfaces, or with all the expansion and therefore the initial condensation taking place in one cylinder. These facts are borne out by practice, though in every type of reciprocating steam engine it has been found advantageous to use superheated steam.

The question of the amount of superheat permissible is bound up with the highest temperature which a cylinder will stand. A good working figure for this value is 500° F. With steam at 175 lb. (abs.) (370° F.) this leaves 130° F. for superheat. There are, however, engines running satisfactorily with steam temperatures as high as 650° F., as can be seen from the tests given on p. 130. The oil used inside the cylinder must be a mineral oil, which does not disintegrate at the highest temperature of the steam, and metallic packing should be used for the steam glands. The phenomenon known as the "growth" of cast iron, referred to on p. 157, should also not be lost sight of. Although so far there are no published cases of trouble traceable to this cause in steam-engine cylinders, cast-iron steam pipes and stop valves have been known to be affected by superheated steam.[1]

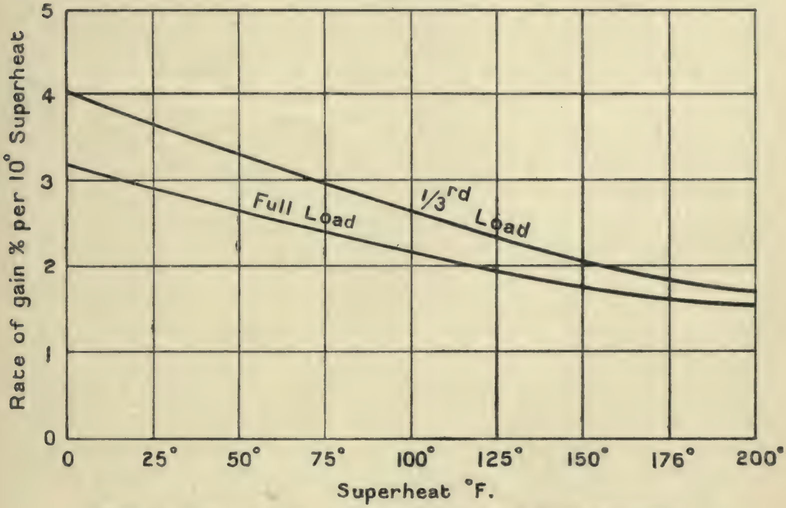


FIG. 82.—Per cent. improvement for every 10° F. of superheat.

(*c*) *Jacketing the Cylinder with Steam* to keep up its mean temperature has the same general effect as using superheated steam. It has been widely used in the past, particularly when trouble was experienced at the first introduction of superheat some fifty years ago. A considerable amount of research work has established the usefulness of steam jackets in keeping down initial condensation, and even in reducing valve leakage by maintaining the face of the valve at a higher average temperature. A number of references may be found in Inchley's "Theory of Heat Engines,"[2] which need not be repeated here, as the tendency in present-day practice is to do without jackets altogether by achieving the same result in a more convenient way. On the other hand, cylinder and steam pipes should always be covered with some form of non-conducting material and lagged with wood or sheet metal.

[1] The whole question is discussed in "Cast Iron in the Light of Recent Research," by W. H. Hatfield, D.Met. 2nd ed. 1918, chap. xii. See also p. 243 in this book.
[2] Second edit., 1920, p. 128.

(*d*) *Expanding the Steam in more than one Cylinder* or compounding has the effect of reducing the cut-off referred to the whole engine, and enables higher boiler pressures to be used. The same missing quantity is spread over more than one cylinder, and is therefore less in each individual cylinder. The temperature range is less in each cylinder, and since initial condensation depends on the *mean* temperature range of the cylinder walls this is reduced. The range of pressure is less in each cylinder, and therefore the valve leakage is less, which again reduces the missing quantity. By suitably arranging the cranks better balancing and also a more even turning moment on the crankshaft can be obtained.

The choice of a suitable ratio of cylinder volumes is affected by a number of factors, most of which are the outcome of practical considerations. For any given set of conditions it is of course possible to calculate the areas of the cylinders, so that both the initial loads on the pistons as well as the work done in each cylinder are approximately equal.[1] This is desirable in every case where there is more than a single crank, but is not so important in the tandem compound type. Other factors which have to be taken into account are the load variations, the type of governing, and the use or otherwise of superheated steam. If the engine has to cope with heavy overloads the normal cut-off in the H.P. cylinder should be early, and this can be obtained by using a relatively large H.P. cylinder, that is, a smaller cylinder ratio than that which would distribute the load or the work evenly between the cylinders.

Cut-off governing on the H.P. cylinder, by increasing the number of expansions in that cylinder as the load decreases, tends to reduce the amount of work left for the L.P. cylinder to do. Throttle governing, by reducing the initial pressure on the H.P. piston, has the opposite effect if the L.P. cylinder and speed of the engine are not altered. As this type of governing is common for high-speed engines, these latter are frequently designed with a smaller cylinder ratio than would be the case if cut-off governing were employed. The effect of using superheated steam is to lower slightly the expansion curve after cut-off, and therefore the mean effective pressure of the indicator diagram. For this reason, possibly, it has been found in practice that a small cylinder ratio again shows to advantage.

Fig. 83 is included here more as a guide for preliminary design than as embodying any hard and fast rule. It shows average cylinder ratios that may be expected with normal conditions if the piston loads and work distribution are kept approximately equal in the different cylinders. It will be found that modifications, as the rusult of such factors as are outlined above, will, in general, tend to increase the size of the high-pressure cylinder, or, in other words, reduce the ratios of the cylinder volumes below those given on the diagram.

(*e*) *Improvement in Mechanical Design.*—From the point of view of achieving economy, mechanical improvements in late years have been chiefly concerned with valve design. The original "D"-slide valve became the piston valve, which was the slide valve balanced all round

[1] See Inchley, "Theory of Heat Engines," 2nd ed. 1920, pp. 145–150, or Unwin, "Elements of Machine Design," 1912, Part ii. pp. 90–92.

the seating. This type of valve, which can be seen in Fig. 71 on p. 129, is still in common use for high-speed engines. Such valves slide to and fro over the steam and exhaust ports, and are usually driven from the crankshaft by means of eccentrics. The same valve controls both the entry and the exit of the steam, which must in consequence pass through the same ports and clearance space at very different temperatures during each stroke. This has the effect of lowering the mean temperature of important parts of the cylinder and adversely affects initial condensation.

The introduction from America of the Corliss type of valve was rapidly recognised as an improvement, particularly for slow-speed

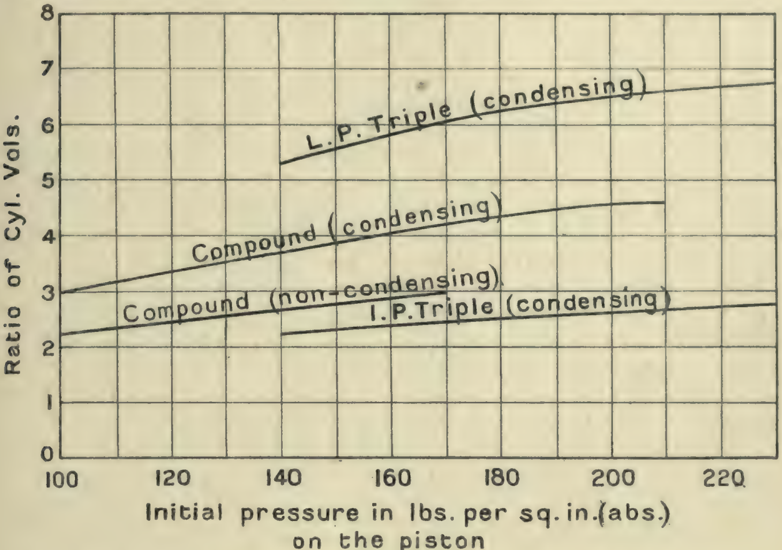


FIG. 83.—Ratios of steam-engine cylinder volumes, referred to H.P. cylinder.

engines of large power, such as are used to drive mills. In this type the inlet and exhaust on both sides of the piston are kept separate, and the cut-off is controlled by a well-designed form of trip action. All four valves are usually operated from a rocking disc driven by a single eccentric. More recently still the mushroom type of valve, or drop valve, as it is usually called, has been introduced by Sulzer Bros., of Switzerland, and achieves much the same results as the Corliss valve in a somewhat neater way. In this type separate valves are also used for inlet and exhaust on each side of the piston, and the valves, which are not unlike gas-engine valves, are driven by a trip action through rods, from eccentrics on a layshaft. An example of such valves may be seen in Fig. 65 on p. 121. It will be noticed that they are made double beat and therefore balanced.

The effect of Corliss and Sulzer valves has been to improve the consumption of those engines for which they are suited, chiefly because the separation of the high-temperature and low-temperature steam

helps to reduce initial condensation. The sharp cut-off attainable with the trip action of such valves also lends itself to more accurate governing and adjustment to different steam conditions.

The cross-sectional area of the ports or steam passages controlled by the valves is usually calculated to give suitable velocities of the steam flowing through them. For common inlet and exhaust ports (condensing) an average value for such steam velocity is 100 ft. per sec., but 120 ft. per sec. may be used with high-speed engines. Non-condensing engines would be lower, say 80 ft. per sec. For Corliss or Sulzer types of valves these figures may be used for the exhaust ports, and 120 to 140 ft. per sec. for steam ports, according to the speed of the engine. Such engines are rarely made non-condensing.

It will be noted that the improvement in economy of the steam engine which follows from compounding or valve design has been obtained, consciously or unconsciously, by the reduction of initial condensation and of the waste which comes from large fluctuations of temperature in the cycle of operations. There is, however, a way of avoiding compounding altogether and of obtaining large powers economically in one cylinder. The method was originally devised by L. J. Todd in 1885, and is opening out a new area in steam-engine design. In patents taken out about that time [1] Todd describes a steam engine with a longer cylinder than the ordinary type, and a piston which fills nearly half the cylinder. Steam is admitted at each end of the cylinder in turn, and exhausts through ports all round the centre of the cylinder, which are uncovered by the piston itself. From the fact that the steam only flows one way in each half of the cylinder and never retraces its path, this type of engine has come to be known as the *Uniflow engine.* Some authorities refer to it as the "Unaflow" engine, and it is also called the "central-exhaust" engine by some makers. Todd in his specifications called it a "terminal-exhaust" engine, and shows by his wording that he was clearly alive to the advantages of such an arrangement.

Very little was heard of the practical application of Todd's principle until some twenty years later, when Professor J. Stumpf, of Charlottenberg, Berlin, published a book [2] on the subject and showed that higher thermal efficiencies and better steam consumptions were possible with uniflow engines than had before been attained with either the simple or compound types of steam engine. The author of that book has done so much towards overcoming the practical difficulties of applying the uniflow principle to the reciprocating steam engine that the type is also known as the Stumpf engine.

In the uniflow engine the two ends of the cylinder come in contact with only high-pressure steam, and therefore remain relatively hot compared to the central portion, which is swept only by the exhaust and therefore comparatively cool steam. The consequence is that fluctuations in temperature of any part of the cylinder are reduced to a minimum and initial condensation is correspondingly decreased. These conditions, whilst being advantageous from a thermal point of view,

[1] British Patent No. 7301 of 1885.

[2] "Die Gleichstrom Dampfmaschine," by J. Stumpf (1911). There is an English translation of this book.

introduce difficulties into manufacture, in that allowance ought to be made in the bore when cold to ensure both the cylinder and the box piston being parallel throughout their length when working. This point was not at first clearly recognised, and led to an unduly large number of cracked cylinders, with a corresponding prejudice against this type of engine. By making suitable assumptions the necessary contraction in the diameter of each end when cold can be calculated. In the absence of specific data this allowance may be taken as $\frac{D}{500}$ for an unjacketed cylinder, and $\frac{d}{1000}$ for the piston. The maximum diameter of the piston must of course not exceed the minimum diameter of the cylinder, so as to prevent any possibility of the piston seizing. The design of the valves and the correct proportion of the valve passages require more care than with older types of steam engines, because the valve has to open and close and the steam has to be introduced over a much shorter portion of the stroke. The adoption of the uniflow principle allows cut-off to take place between $\frac{1}{10}$ and $\frac{1}{3}$ in a single cylinder. The indicator card has all the appearance of compound- or triple-expansion cards when reduced to the L.P. cylinder, with the absence of any loss of work between the cards, that is, in intermediate receivers or steam chests.

This can be seen in the accompanying diagrams, Fig. 84 being reproduced from Inchley's "Theory of Heat Engines," and Fig. 85 from one of the few papers yet devoted to the uniflow steam engine,[1] read by H. Pilling before the Manchester Association of Engineers in 1920. In this paper Mr. Pilling, an authority on the manufacture of uniflow engines, points out that this type is most suitable where condensing water is available and assured, and where the direction of rotation is continuous; that is to say, for driving mills and generating electric power. The heavy box piston does not lend itself to high speeds or rapid reversing, although the uniflow has been tried in locomotive design.

There is one point which has to be carefully watched in connection with uniflow engines. Since compression starts as soon as the exhaust ports are covered by the piston the pressures reached at the end of the compression stroke may become abnormal, unless some form of relief valve is fitted near the cylinder head. In practice the exhaust ports are made $\frac{1}{10}$ of the stroke in length, which means that compression is taking place during $\frac{9}{10}$ of the stroke. The indicator card shown in Fig. 85 was taken with a vacuum of 27 in. or $1\frac{1}{2}$ lb. back pressure, and the compression reaches about 130 lb. per sq. in. (abs.), the clearance in this case being 3 per cent. Since the compression curve follows the law of $pv^n =$ constant, the final pressure will vary directly as the back pressure, so that if the vacuum failed and the back pressure became 15 lb. per sq. in. the compression would reach $130 \times \frac{5}{1 \cdot 5}$ $= 1300$ lb. per sq. in. It is possible to design non-condensing

[1] There is another paper by F. B. Perry on "The Uniflow Steam Engine," in *Proc. Inst. Mech. Eng.*, July, 1920.

uniflow engines, provided special precautions are taken, but the type is more common in America, where the majority of steam-plants are made non-condensing.

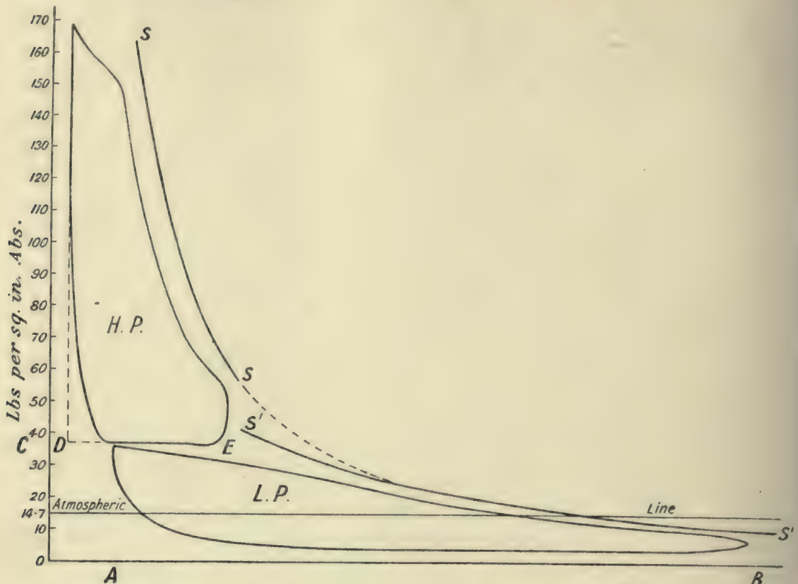


FIG. 84.—Compound engine indicator cards reduced to L.P. cylinder.

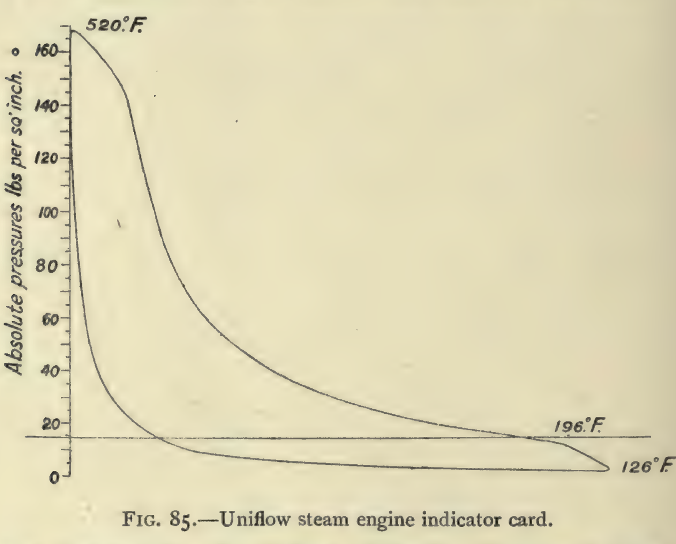


FIG. 85.—Uniflow steam engine indicator card.

The Principles of Steam-Engine Design.—There are numerous ways of approaching the preliminary design of a new steam engine for any given purpose, and the method indicated here must only

be taken as representative of general principles. It is usual to ascertain first what conditions are laid down, or to decide on suitable conditions if none are given. These conditions include: power of the engine; steam conditions, such as available steam pressure, superheat, condensing or non-condensing; speed of engine (if fixed); and, finally, type of engine. Very frequently the first two of these conditions are fixed, whilst the remaining two are left to the discretion of the designer.

Theoretical Indicator Cards and the Diagram Factor.—Whatever method of design is proposed, it cannot fail to be of value to attempt to forecast the behaviour of the steam in the cylinders by drawing from a theoretical standpoint an indicator card which conforms with the conditions laid down or assumed. It is, of course, better still if actual cards are available, which have been taken from engines of approximately the same type, power, and speed, working under exactly the same conditions. But in the absence of such information it is still possible to draw a reasonably accurate card if sufficient data is available. In practice this is hardly ever done when the only point that is required is the *mean effective pressure* (M.E.P.) from which to calculate the dimensions of the cylinders. In that case either a purely ideal card is used or more commonly a formula based on such a card. If, however, a diagram is required to estimate the steam consumption or with which to combine the inertia effects of the reciprocating parts in order to arrive at a turning-moment diagram for calculating the dimensions of the flywheel, then it is desirable, although more laborious, to forecast the actual indicator card more accurately.

Taking the purely ideal considerations first, there are various ways in which an ideal card may be drawn for any given conditions, from which to calculate an ideal mean effective pressure. This M.E.P., when multiplied by a suitable fraction known as the *diagram factor*, gives the actual M.E.P. which may be expected under working conditions. The usual formula for the ideal M.E.P. is based on hyperbolic expansion (pv = constant) and neglects clearance as well as back pressure.

$$\text{M.E.P.} = \frac{p_1}{r}(1 + \log_e r) \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where p_1 = initial steam pressure in lb. per sq. in. (abs.)

r = apparent ratio of expansion, usually taken as the reciprocal of the cut-off.

In the case of more than one cylinder the cards are combined as in Fig. 84, and the M.E.P. and r are referred to the low pressure cylinder.

If the back pressure (p_b) is known or assumed

$$\text{M.E.P.} = \frac{p_1}{r}(1 + \log_e r) - p_b \quad . \quad . \quad . \quad . \quad . \quad (2)$$

If in addition a clearance volume of c times the piston displacement is allowed for

$$\text{M.E.P.} = \frac{p_1}{r}\left\{1 + (1 + cr)\log_e \frac{1 + c}{\frac{1}{r} + c}\right\} - p_b \quad . \quad . \quad (3)$$

The selection of a suitable diagram factor requires discretion. It depends in the first place on whether it is compared with the conditions shown in formulas (1), (2), or (3), and care should be exercised in ascertaining which has been used in any given value.

In addition, the diagram factor may be affected by the value of r, the ratio of expansion; the initial pressure (p_1); the state of the steam, *i.e.* whether superheated or not; the value of the back pressure (p_2), *i.e.* whether condensing or non-condensing; the speed of the engine; and the type of engine and valves, *e.g.* simple, compound, or uniflow, jacketed or not jacketed.[1] In the absence of more detailed information the values given in the following table will be found to give a design in accordance with modern practice. The diagram factor (k) is referred to formula (2). For convenience in calculations our table also shows—

p_1 = initial pressure on piston in lb. per sq. in. (abs.)
r = apparent ratio of expansion at normal full load, and
c = clearance volume as a proportion of the stroke
both r, and c being referred to L.P. cylinder in the case of compound engines
s = average mean piston speed in ft. per minute
η = mechanical efficiency $\cdot \dfrac{\text{B.H.P.}}{\text{I.H.P.}}$

Type of engine.	p_1	r	c	s	η	k
A. *Horizontal engines*—						
(1) *simple* (slide-valve)—						
non-condensing	100	2	0·12	500	85	0·80
condensing	80	4	0·10	500	85	0·75
(2) *compound* condensing—						
(Corliss or drop-valve) . . .	165	12	0·05	600	90	0·70
ditto with superheat . . .	165	12	0·05	600	90	0·65
(3) *uniflow* condensing—						
(drop-valve)	165	10–20	0·03	750	88	0·76
ditto with superheat . . .	165	10–20	0·03	750	88	0·71
B. *Vertical engines*—						
(4) *simple* condensing	100	3	—	600	88	0·70
(5) *compound* condensing with superheat	165	12	—	750	90	0·60
(6) *triple expansion* condensing with superheat	220	16	—	750	92	0·55
(7) *marine* condensing—						
compound	—	8	—	—	—	0·70
triple-expansion	—	12	—	—	—	0·625

The actual indicator diagram in nearly every case falls inside the ideal card just considered. Owing to the area of the cylinder being so much larger than the cross-sectional area of the inlet port the steam cannot enter quickly enough to maintain full initial pressure up to the time of closing the valve. This effect, which is known as *wire-drawing*, results in the pressure at the point of cut-off, and therefore at the point where expansion begins, being less than the initial pressure on the

[1] For a discussion on the effect of these factors, see Inchley's "Theory of Heat Engines," 2nd ed. 1920, pp. 131 *et seq.*; also Unwin's "Machine Design," 1912, Part ii. pp. 79-88.

piston. Information on this point can be obtained by the study of actual indicator cards taken with different speeds, types of valves, points of cut-off, and steam conditions. For H.P. cylinders and a cut-off in the neighbourhood of $\frac{1}{4}$ this pressure will be found to be about 85 per cent. of the *range* of pressure represented on the card (p_1-p_2). For a cut-off approximating to $\frac{1}{2}$ this value may be as low as 65 per cent. For L.P. cylinders in which the ratio of areas is usually larger the corresponding values may be 75 per cent. with $\frac{1}{4}$ cut-off to 55 per cent. with cut-off about $\frac{1}{2}$. For uniflow engines it may be taken at 88 per cent. with $\frac{1}{10}$ or less cut-off, and 75 per cent. with $\frac{1}{3}$ cut-off.

Piston and slide-valves which close gradually will always give a rounded corner at cut-off, but it is possible with drop-valves, and to a slightly less extent with Corliss valves, to obtain a very clean cut-off with hardly any wire-drawing effect as the valve closes.

The expansion curve in the ideal card corresponds to $pv^{1·0}$ = constant. The actual curve immediately after cut-off will lie below this; that is, it will follow the law of pv^n = constant, where n is greater than 1. The value of n may be taken as 1·1 for steam which is not superheated, and as 1·2 for steam with an initial superheat of 100° F. or more. The expansion curve should not be taken lower than 8 lb. per sq. in. (abs.) for condensing or 20 lb. per sq. in. (abs.) non-condensing. At this point, which is known as the *terminal pressure*, release takes place and the toe of the diagram is rounded off to the end of the stroke.

In order that the steam may leave the cylinder the exhaust should start at a back pressure which is 2 to 3 lb. higher than the pressure in the condenser or of the atmosphere, as the case may be. In uniflow engines with their much larger exhaust area the back pressure need not be more than 0·5 lb. above that of the condenser. Some makers assume that there is no difference between back pressure and vacuum at the toe of the diagram, but this would theoretically prevent a flow from the cylinder to the condenser, and it is better to allow a small margin of back pressure. The exhaust valve opens gradually, and the diagram toe should be again rounded off underneath to run into the straight lines corresponding to this cylinder back pressure. As the exhaust proceeds the back pressure may be taken to drop evenly to the vacuum pressure at the point where compression commences.

The compression curve approximates to the law of $pv^{1·2}$ = constant, and commences when the exhaust valve closes. This may be anywhere between 0·7 and 0·9 of the return stroke or even later. For uniflow engines compression starts much earlier, and may be taken at 0·1 of the return stroke. The compression curve becomes a straight line again as it runs up towards the point of admission. This latter takes place just before the end of the stroke, and the type of valve again determines the shape of this corner of the indicator card. Such a diagram, if drawn with due regard to all the conditions under which the given engine is expected to work, should enable the designer to determine the M.E.P. without recourse to a diagram factor, the distribution of pressure throughout the stroke and the probable steam consumption as far as it can be ascertained from an indicator card.

Example of Steam Engine Preliminary Design.—To illustrate these various points it may be of help to work through an example

of preliminary design, in which the power of the engine and the steam conditions are given, but which leaves the choice of speed and the type of engine to the discretion of the designer. In this example a textile mill driven by ropes requires a steam engine suitable for developing 400 horse-power under the following steam conditions: 160 lb. per sq. in. (gauge) at the stop-valve, 100° F. superheat available, condenser plant capable of maintaining 26 in. vacuum (Bar. 30).

Choice of Type and Speed.—A visit to the power houses of textile mills in Lancashire and Yorkshire will show a surprising variety of steam engines still in daily use. Examples of beam engines driving through cast-iron gearing were still to be seen running in 1920. For a long time horizontal tandem compounds with Corliss valves were considered most reliable and economical, but these are gradually being replaced by the cross-compound type with drop-valves. This may be looked upon as the most representative steam engine for slow-speed steady drive at the present time (1921), though the uniflow engine must be considered a step in advance, and may some day replace it in mills that retain rope drive or that require steam for other purposes. This design will therefore be worked out for a horizontal cross-compound (H.C.C.) with drop-valves, and as an alternative for a uniflow (U) with drop-valves. The speeds of these engines may be taken as those which conform to current practice for the size under consideration, say 120 R.P.M. for the H.C.C. and 150 R.P.M. for the U.

Determination of Cylinder Dimensions.—The proportions of the cylinders may be provisionally determined with the help of the table given on p. 180.

	H.C.C.	Uniflow.	Unit.
I.H.P. $= \frac{\text{B.H.P.}}{\eta}$	$\frac{400}{0{\cdot}90} = 444$	$\frac{400}{0{\cdot}88} = 453$	—
r (apparent ratio of expansion)	12	10	—
p_1 (initial piston pressure)	$160 + 15 - 10 = 165$	165	lb. per sq. in. (abs.)
(Ideal) p_b (for 26 in. vac.)	2	2	lb. per sq. in. (abs.)
M.E.P. p_m (from formula 2)	$\frac{165}{12}(1 + 2{\cdot}3 \log 12) - 2$ $= 47{\cdot}5 - 2$ $= 45{\cdot}5$	$\frac{165}{10}(1 + 2{\cdot}3 \log 10) - 2$ $= 54{\cdot}3 - 2$ $= 52{\cdot}2$	lb. per sq. in. (abs.)
k (diagram factor) with superheat	0·65	0·71	—
Actual M.E.P. p_m	$45{\cdot}5 \times 0{\cdot}65 = 29{\cdot}6$	$52{\cdot}3 \times 0{\cdot}71 = 37{\cdot}25$	lb. per sq. in. (abs.)
s (piston speed) = 2LN	600	750	ft. per min.
Area of L.P. cyl. $= \frac{\text{I.H.P.} \times 33{,}000}{2\text{LN} \times \text{M.E.P.}}$	$\frac{444 \times 33{,}000}{600 \times 29{\cdot}6} = 825$	$\frac{453 \times 33{,}000}{750 \times 37{\cdot}25} = 536$	sq. in.
Dia. of L.P. cyl. (to nearest $\frac{1}{8}$ in.)	$32\frac{3}{8}$	$26\frac{1}{8}$	in.
Cyl. vol. ratio	4 to 1	—	—
Dia. of H.P. cyl.	$16\frac{3}{16}$	—	in.
Stroke (L) $= \frac{s}{2\text{N}}$	$\frac{600}{2 \times 120} = 2{\cdot}5$	$\frac{750}{2 \times 150} = 2{\cdot}5$	ft.

A manufacturer, particularly if he is building a series of similar engines for different powers, would probably round off the dimensions of his cylinders, but for the purposes of this illustration they may be left as calculated.

At this point the compound engine may be conveniently checked against an ideal indicator card for initial piston loads and distribution of work, and the cut-off in the two cylinders provisionally determined.

Neglecting clearance and assuming hyperbolic expansion for equal initial loads on the pistons

$$p_1 - p_r = n(p_r - p_b)$$

where p_1 = initial pressure in lb. per sq. in. (abs.)
p_r = receiver pressure in lb. per sq. in. (abs.)
p_b = back pressure (at opening of exhaust)
n = ratio of cylinder volumes

In the example—

$$165 - p_r = 4(p_r - 4)$$
$$5p_r = 165 + 16$$
$$p_r = \tfrac{181}{5} = 36 \text{ lb. per sq. in. (abs.)}$$

Let r = total ratio of expansion (apparent)
r_h = ratio of expansion in H.P. cylinder
r_l = ratio of expansion in L.P. cylinder

then, still neglecting clearance,

the cut-off in the H.P. cylinder

$$= \frac{n}{r} = \frac{4}{12} = \frac{1}{3} = \frac{1}{r_h}$$

and the terminal pressure in H.P. cylinder

$$= \frac{p_1}{r_h} = \frac{165}{3} = 55 \text{ lb. per sq. in. (abs.)}$$

Again, the terminal pressure (p_t) in L.P. cylinder

$$= \frac{p_1}{r} = \frac{165}{12} = 13{\cdot}8 \text{ lb. per sq. in. (abs.)}$$

and the cut-off in L.P. cylinder will therefore be

$$\frac{p_t}{p_r} = \frac{13{\cdot}8}{36} = \frac{1}{2{\cdot}625} \quad \text{or} \quad \frac{8}{21}$$

Using formula (2) on p. 179

$$\text{M.E.P. in H.P. cylinder} = \frac{165}{3}(1 + 2{\cdot}3 \log 3) - 36$$
$$= \frac{165}{3}(1 + 1{\cdot}1) - 36$$
$$= 115 - 36 = 79 \text{ lb. per sq. in. (abs.)}$$

$$\text{M.E.P. in L.P. cylinder} = \frac{36}{2{\cdot}625}(1 + 2{\cdot}3 \log 2{\cdot}625) - 4$$
$$= \frac{36}{2{\cdot}625}(1 + 0{\cdot}96) - 4$$
$$= 27 - 4 = 23 \text{ lb. per sq. in. (abs.).}$$

So that by this approximate method—

$$\frac{\text{work done in H.P. cylinder}}{\text{work done in L.P. cylinder}} = \frac{79}{4 \times 23} = \frac{79}{92} = \frac{1}{1 \cdot 16}$$

$$\text{cut off in H.P. cylinder} = \frac{1}{3}$$

$$\text{cut off in L.P. cylinder} = \frac{1}{2 \cdot 625}$$

$$\text{receiver pressure} = 36 \text{ lb. per sq. in. (abs.)}$$

Construction of Actual Indicator Cards.—The methods outlined on p. 181 enable the probable appearance of the actual indicator cards of

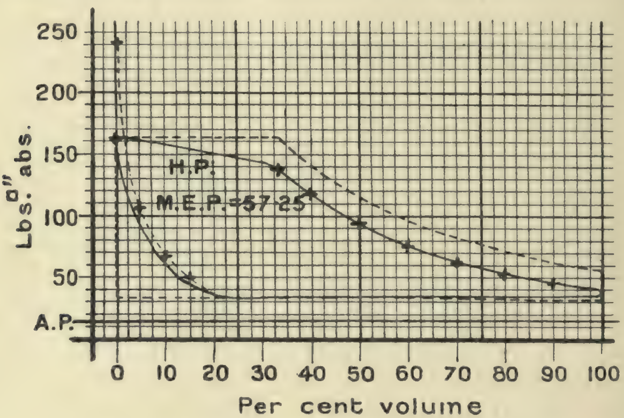


FIG. 86.—Constructed indicator card for H.P. cylinder.

these engines to be determined when working at normal full load. Figs. 86 and 87 are the cards for the H.P. and L.P. cylinders of the horizontal cross-compound, and Fig. 88 the card for the uniflow engine.

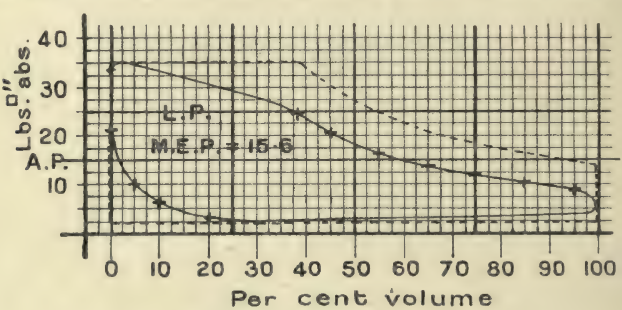


FIG. 87.—Constructed indicator card for L.P. cylinder.

It is a convenience to use squared paper divided to suit the probable scales of the indicator springs. Any convenient horizontal length is chosen for the volume and divided into 100 equal parts. The clearance space (obtained from the table on p. 180) is set back from

zero-volume and a vertical line drawn, upon which the pressure scale is marked from zero in lb. per sq. in. (abs.).

The initial pressure on the piston is then put in on the zero-volume line and the pressure at the point of cut-off calculated from the range of pressure in each card. For the H.C.C. the H.P. cylinder will exhaust about 2 lb. above receiver pressure or $36 + 2 = 38$ lb., so that the pressure range is $165 - 38 = 127$ lb. With cut-off of $\frac{1}{3}$ the drop will be to about $0{\cdot}8 \times 127 = 102$ lb. (see p. 181), to which must be added the back pressure to get $102 + 38 = 140$ lb. per sq. in. on the absolute scale. This point is marked at 33·3 on the volume scale, and joined by a straight line to the initial pressure (Fig. 86).

In a similar way the initial pressure on the L.P. cylinder piston will

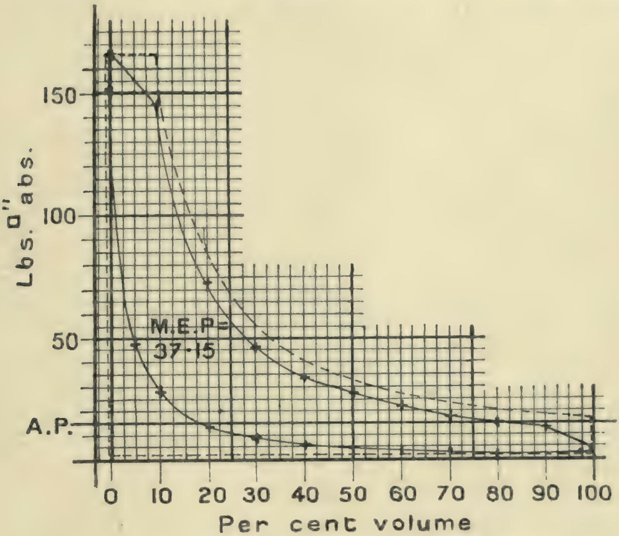


FIG. 88.—Constructed indicator card for uniflow cylinder.

be about 1 lb. below receiver pressure or 35 lb., the back pressure 2 lb. above vacuum or 4 lb., and the wire-drawing factor about 0·65, so that cut-off pressure in L.P. cylinder $= 0{\cdot}65(35 - 4) + 4 = 20{\cdot}2 + 4 = 24{\cdot}2$ lb. per sq. in. (abs.) and the cut-off of $\frac{1}{2{\cdot}625}$ marked at 38·2 on the volume scale (Fig. 87).

For the uniflow the back pressure is taken in this example as 0·5 lb. above vacuum. With a wire-drawing factor of 0·88 the cut-off pressure $= 0{\cdot}88(165 - 2{\cdot}5) + 2{\cdot}5 = 145$ lb. per sq. in. (abs.) at the cut-off 10 on the volume scale (Fig. 88).

These three pressure points, 140 for the H.P. cylinder, 24·2 for the L.P. cylinder, and 145 for the uniflow, give the values which are to be assigned to p in the equation $pv^n =$ constant, which is used to determine the expansion curve. The volume (v) should include the apparent ratio of expansion and the clearance space in each case.

For H.P. cylinder therefore $v = 33{\cdot}5 + 5 = 38{\cdot}3$
For L.P. cylinder „ $v = 38{\cdot}2 + 5 = 43{\cdot}2$
For uniflow „ $v = 10 + 3 = 13$

Since in the example the steam is superheated to 100° F. the value of the index n is taken at 1·2 for the H.P. cylinder and for the uniflow. But by the time the L.P. cylinder is reached very little if any superheat will remain, and n will be represented more correctly by 1·1.

Since $\log p + n \log v = \log \text{constant}$
for H.P. cylinder $\log 140 + 1{\cdot}2 \log 38{\cdot}3 = \log \text{constant}$
or $2{\cdot}1461 + 1{\cdot}2 \times 1{\cdot}5832 = \text{constant}$
whence constant (as a log) $= 4{\cdot}0459$

for a volume of 40 + 5

$1{\cdot}2 \log 45 = 1{\cdot}2 \times 1{\cdot}6532 = 1{\cdot}9838$
hence $\log p_{40} = 4{\cdot}0459 - 1{\cdot}9838 = 2{\cdot}0621$
whence $p_{40} = 115{\cdot}3$ lb. per sq. in. (abs.)

In this way points have been taken for every ten divisions of the volume scale and the corresponding pressure points computed.

For the L.P. cylinder the expression $\log 24{\cdot}2 + 1{\cdot}1 \log 43{\cdot}2$ gives the constant (in the form of a log) as 3·1819, while for the uniflow, $\log 145 + 1{\cdot}2 \log 13 = 3{\cdot}4981$. Pressure points have been calculated for these two cards in exactly the same way as for the H.P. cylinder.

In both the cards of the H.C.C. the toe of the diagram is then rounded off to back pressure, that is, to 38 lb. for H.P. cylinder and to 4 lb. for L.P. cylinder, care being taken in the latter case to release not lower than 8 lb. per sq. in. (abs.). In the uniflow the exhaust is known to open at 90, at which point the pressure will either fall away to back pressure at 100, or drop more gradually and only reach back pressure of 2·5 lb. (abs.) at 90 on the return stroke. The particular form of the toe of the diagram depends on the design of the exhaust ports and varies with different makers.

The next point to be determined is the closing of the exhaust ports. In these cards the following values have been first adopted :—

H.P. cylinder 0·8 of return stroke
L.P. cylinder 0·7 „ „
Uniflow 0·1 „ „

The pressure falls in a straight line to receiver or vacuum pressure, as the case may be, that is, to 36 lb. in H.P. cylinder, and 2 lb. in L.P. cylinder. As mentioned above, some designers of uniflow engines make the pressure drop to about 2·5 lb. at the end of the stroke, and to 2 lb. (vacuum pressure) at the commencement of compression, but it is better to assume a slight difference of pressure between cylinder and condenser to ensure a clear exhaust up to closure.

The compression curves are computed in a similar way to the expansion curves, the expressions being—

H.P. cylinder $\log 36 + 1{\cdot}2 \log 25 = 3{\cdot}2338$
L.P. cylinder $\log 2 + 1{\cdot}2 \log 35 = 2{\cdot}1539$
Uniflow $\log 2{\cdot}5 + 1{\cdot}2 \log 93 = 2{\cdot}7601$

These curves should run into the zero-volume line to join the initial pressure points with which the diagram started.

If, as in the case of the H.P. cylinder here, the compression curve has been started too early it should be recalculated for a later closing of the exhaust valve. In Fig. 86 the first attempt is shown dotted and a satisfactory solution added in full lines.

The ideal *pv* diagram, according to formula (2), has been added in dotted lines on each figure for the sake of comparison. The diagram factor will be the ratio of the area of the actual card to this ideal card or, what amounts to the same thing, the ratio of the mean effective pressures obtained from the two cards. In the case of the compound engine the H.P. cylinder card should either be reduced to the same scale as the L.P. cylinder card in a similar way to the example shown in Fig. 84 on p. 178, or the M.E.P. of the H.P. cylinder divided by the ratio of cylinder volumes and added to the M.E.P. of the L.P. cylinder. The cards as drawn show the following diagram factors :—

$$\text{H.C.C.} = \frac{\frac{56 \cdot 25}{4} + 15 \cdot 6}{45 \cdot 5} = \frac{29 \cdot 6}{45 \cdot 5} = 0 \cdot 65$$

$$\text{Uniflow} = \frac{37 \cdot 15}{52 \cdot 3} = 0 \cdot 71$$

which agree with 0·65 and 0·71 used in the calculations.

SECTION III

INTERNAL-COMBUSTION ENGINES

CHAPTER X

GENERAL CONSIDERATIONS AND THERMAL EFFICIENCY

THE expression *Internal-combustion Engine* refers to any kind of prime mover where the burning of the fuel takes place inside the working cylinder itself. A gun is an internal-combustion machine, and if gunpowder or gun-cotton were as cheap as coal or gas a solid fuel internal-combustion engine would no doubt be constructed. At the present time the fuels in use are limited to liquids and gases, and the two main lines upon which internal-combustion engines have developed are broadly distinguished by the fact that one works on a liquid fuel, such as oil or petrol, whilst the other uses gases, such as are obtained from gasworks, coke-ovens, blast-furnaces, or specially constructed power-gas producers.

It is true that steam in a steam prime mover behaves in many ways like a gas, but its heat has been obtained from an *external* source, such as a steam boiler, and it is not nowadays generated in the cylinder where the work takes place.

As it is the vapour of the liquid fuel that is used in oil engines, both types of internal-combustion engines really work with gaseous mixtures, and their mechanical design, except the part which deals with the fuel, need not differ in essentials. It is possible to use the same theory and to devise the same ideal efficiency to cover both gas and oil engines.

The difficulties in the way of applying thermodynamics to internal-combustion engines are, however, much greater than those met with when steam is considered, particularly in steam turbines. There has been a large amount of valuable work done of an analytical and an experimental nature, but so far it cannot be said that exact knowledge has been obtained of the state of the working fluid inside an internal-combustion cylinder at all parts of the stroke. That this should be so may be readily recognised when it is realised that the working fuel suitable for internal combustion may vary in both its chemical and physical properties and in the strength of its mixture with the air required for combustion. Again, in all modern types the working fluid is ignited under compression, and the amount of this compression introduces another variable.

In its briefest form, the problem to be solved consists in predicting the thermodynamical relations when an explosive mixture of air and gas or vaporised oil is introduced into a vessel with a movable piston; compressed either before or after the introduction of the combustible as distinct from the air; ignited; expanded in doing work; and the spent products of combustion removed from the cylinder before the introduction of a fresh charge.

The way in which this removal takes place distinguish the "four-stroke" cycle from the "two-stroke" cycle. In the former, which is frequently known as the Otto cycle, the piston, after the working stroke, acts as a pump for three strokes, which means that there is one working stroke in every two revolutions of the engine. The piston for these three strokes is driven by the energy stored up in the flywheel, though of course by arranging a number of cylinders on one crank, firing at different times, the load can be distributed between the several cylinders. On the first return stroke of the piston the products of combustion are pushed out through exhaust valves situated at the head of the cylinder. On the next forward stroke the piston draws in a fresh charge through suitably operated inlet valves, and another second return stroke compresses this charge to be fired on the working stroke.

In the two-stroke or Clerk cycle arrangements are made for removing the products of combustion at the end of the exhaust stroke, and introducing the fresh charge for the piston to compress on its first return stroke. There is therefore one working stroke per cylinder for each revolution of the engine.

The different ways in which both the four-stroke and the two-stroke order of events can be carried out are usually known as *cycles of operation*, and distinction is made as to whether combustion takes place (*i.e.* heat is received) at—

1. Constant volume.
2. Constant pressure.
3. Both constant volume and constant pressure.
4. Constant temperature.

The fourth type represents the *Carnot cycle*, which is not used in internal-combustion engines, but which forms an unattainable ideal when steam or hot air are considered. The constant-volume cycle is the one most commonly used for both gas and oil engines, and the majority of internal-combustion engines approximate to this type. The appearance of the indicator card of this cycle is familiar to all students of thermodynamics, and is characterised by a pointed top or peak more or less vertical above the point of compression volume. This indicates that the volume of the working fluid has remained (more or less) constant during combustion whilst the pressure rises to a maximum, but as soon as burning has ceased the pressure drops and the products of combustion commence expanding. In the large majority of engines using this cycle expansion is not carried further than the volume at which compression commences, which entails a vertical drop in pressure at the "toe" of the diagram. The cycle in this form is the one used by the Committee of the Institution of Civil Engineers, referred to below, to deduce the ideal standard of efficiency which is shown to cover the constant-pressure type and the constant-

temperature type as well. One or two engines, notably the Atkinson engine of 1885, and more recently the Humphrey gas pump (1909), approximate more nearly to complete expansion down to the pressure at which compression commences on the return stroke.

In the constant-pressure type compression is carried right up to the maximum pressure, and the mixture burns with expanding volume but with little or no drop in pressure. This accounts for the flat top of the indicator cards which characterise the Diesel oil engine, and has given rise to the term "slow-combustion" which is sometimes applied to this engine. After firing, the pressure decreases as the volume expands down to compression pressure. In the Diesel engine, which is the best known example of this cycle, expansion is not complete, and the pressure drops vertically at the "toe" of the diagram without further increase in volume.

The third type, which is sometimes called the *Dual-combustion* cycle, has not received the attention it deserves. It represents the case of an internal-combustion engine in which either the fuel is injected in two stages, or ignition takes place at two points of the working cycle, so that heat is received at both constant volume and constant pressure. The characteristic card of this type would show both a vertical line from the end of compression up to maximum pressure and a flat top.[1] A few oil engines approximate to this cycle, such as the Sabathé in France, the De la Vergne[2] in America, and the Blackstone hot-bulb oil engine in England. A careful examination of the "cold-starting" or "solid-injection" type of oil engines shows that most of them also approximate to this cycle.

Ideal Thermal Efficiencies with Constant Specific Heats.—The first three cycles in the above list were thoroughly analysed by a Committee of the Institution of Civil Engineers, who were appointed in 1903 to report on the standards of efficiency of internal-combustion engines.[3] The reports of this committee form a valuable landmark in the application of thermodynamics to practice in the design of internal-combustion engines. They contain a summary of experimental and theoretical work bearing on the subject to date. It is shown that the ideal efficiency of all three cycles can be expressed in one formula if the following assumptions are made :—

(1) The reception and rejection of heat should take place as nearly as may be in the same way as in an actual engine.

(2) There should be no heat losses due to conduction, radiation, leakage, or imperfect combustion.

(3) There is no change in the specific heat of the working substance at high temperatures or pressures.

[1] See *Proc. Inst. Mech. Eng.* March, 1912, p. 234, for an example of this type of indicator card which was obtained by Sir Dugald Clerk, and given by him in the discussion on R. Diesel's paper on "The Diesel Engine."

[2] See paper entitled "Symposium on Oil Engines," by H. R. Setz, *Trans. Am. Soc. Mech. Eng.* vol. xxxiii. (1911), pp. 870 and 878.

[3] Preliminary Report, published in *Proc. Inst. C.E.* vol. clxii. (1905), pp. 307–338. Final Report, containing a description of gas-engine trials, *Proc. Inst. C.E.* vol. clxiii. pp. 241–288. For a full account, see "The Gas, Petrol, and Oil Engine," by Sir Dugald Clerk, vol. i. 1910 ed.

(4) The constant-volume cycle only expands down to the volume at which compression starts, which involves with (approximately) adiabatic curves a vertical drop in pressure at the "toe" of the diagram.

(5) The constant-pressure (and constant-temperature) cycles on the other hand expand down to the same pressure at which compression starts.

(6) The working fluid is considered to be air.

The common expression, deduced under these conditions, is known as the *air standard of efficiency*, and is given by the formula

$$\eta_v = 1 - \left(\frac{1}{r}\right)^{\gamma - 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where η_v = the ideal thermal efficiency (constant volume)

r = the ratio of compression

γ = the ratio of the specific heat of the working fluid at constant pressure to the specific heat at constant volume (assumed constant)

$= \frac{K_p}{K_v}$ at N.T.P. = 1·4 for air

The actual value of this air standard efficiency for any compression ratio found in practice can be read off direct from Fig. 89. The scale,

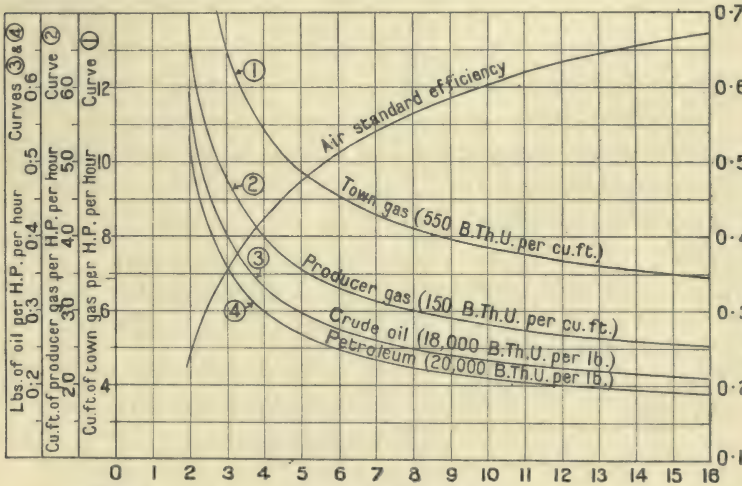


FIG. 89.—Ideal consumption curves for I.C. engines.

which is on the right-hand side of the diagram, should be multiplied by 100 if its value is required as a percentage.

The Committee of the Institution of Civil Engineers, whilst clearly pointing out on pp. 317 and 331 of their report the limitations of this standard, recommended its adoption as a basis of comparison for all

types of internal-combustion engines, and it is at present used in much the same way as the Rankine cycle for steam engines and turbines.

Fig. 89 also shows ideal consumptions on the air standard of efficiency for average values of fuels used in internal-combustion engines. It should be noted that the two curves for gas are drawn to different scales, shown on the left-hand side of the diagram, and that a third scale has been used for the two oil curves drawn with full lines. The way these curves are obtained may be illustrated by means of an example.

A large Diesel engine (working on the four-stroke cycle) has a compression ratio (r) of 12. The efficiency curve shows that the corresponding air standard would be 0·635. If the engine uses crude oil with a lower calorific value of 18,000 B.Th.U. per lb., the ideal could only utilise $0{\cdot}635 \times 18{,}000 = 11{,}420$ B.Th.U. in each lb. of oil. As has already been shown on p. 97, the number of B.Th.U. corresponding to one horse-power per hour is

$$\frac{33{,}000}{778} \times 60 = 2546 \text{ B.Th.U.}$$

Therefore the number of lb. of oil per H.P. per hour required by the engine working ideally

$$= \frac{2546}{11{,}420} = 0{\cdot}223$$

which corresponds to the point on curve (3) for a compression ratio of 12.

With the aid of such curves as these it is possible to find out at once how near any particular engine is approaching the ideal condition of the air standard, that is to say, what its coefficient of performance is. The Diesel engine just referred to had an actual consumption of 0·38 lb. of oil per B.H.P. per hour at full load when using crude oil of a lower calorific value of 18,000 B.Th.U. per lb. The coefficient of performance would therefore be $\frac{0{\cdot}223}{0{\cdot}38} = 0{\cdot}587$.

Some such method of considering results actually obtained in practice gives a much clearer idea to the designer how near the ideal is reached in any particular engine than the more commercial method of working out the absolute thermal efficiency. This, as in the case of steam, equals

$$\frac{\text{heat turned into work}}{\text{total heat available in the fuel}}$$

In the Diesel engine considered above the total heat available in the fuel per B.H.P. per hour $= 0{\cdot}38 \times 18{,}000 = 6840$ B.Th.U., and the heat turned into work = 2546 B.Th.U. as before. The absolute thermal efficiency of this Diesel engine would therefore be

$$\tfrac{2546}{6840} = 0{\cdot}372, \text{ or } 37{\cdot}2 \text{ per cent.}$$

When the air standard of efficiency was recommended by the 1905 Committee of the Institution of Civil Engineers, the report was careful to point out that the specific heat of gases at the high temperatures and pressures reached in internal-combustion engines was not constant but

undoubtedly varied to an extent which would materially effect the results. The knowledge of accurate values at the time the report was issued was confined to specific heat at ordinary temperatures, so that until further information was available it was deemed advisable to ignore this variation.

The ratio of the efficiency of any particular internal-combustion engine when calculated with constant specific heat to the efficiency when variable specific heat is allowed for is so nearly constant that the air standard gives more or less accurate *relative* results when used for the constant-volume cycle, which conforms most nearly to what actually occurs in explosion engines. The same thing may be said of the constant-pressure type (Diesel engine) if the air standard efficiency is modified as follows, to allow for the fact that the toe of the diagram shows incomplete expansion. The efficiency, assuming constant specific heat, for the Diesel engine may then be written

$$\eta_p = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{\rho^\gamma - 1}{\gamma(\rho - 1)} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where η_p = ideal efficiency (constant pressure)

$$r = \frac{\text{cylinder volume before compression}}{\text{cylinder volume after compression}}$$

$$\rho = \frac{\text{cylinder volume at ``cut off''}}{\text{cylinder volume after compression}}$$

$$\gamma = \frac{\text{specific heat at constant pressure } (K_p)}{\text{specific heat at constant volume } (K_v)}$$

The expression "cut off" by analogy with the steam-engine indicator card represents the point where constant pressure ceases and the gas continues expanding with a falling pressure according to the law pv^n = constant. Since the term $\frac{\rho^\gamma - 1}{\gamma(\rho - 1)}$ is greater than unity the Diesel efficiency will be less than that deduced by the air standard, and the coefficient of performance therefore greater than when the air standard is used as in the example just given.

The corresponding efficiency for the dual-combustion cycle with constant specific heats is shown by W. J. Walker[1] to be

$$\eta_m = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \left\{\frac{\alpha\rho^\gamma - 1}{\alpha - 1 + \gamma\alpha(\rho - 1)}\right\} \quad . \quad . \quad . \quad (3)$$

where $\alpha = \dfrac{\text{maximum pressure after firing}}{\text{pressure at end of compression before firing}}$

η_m = ideal efficiency (dual combustion)

and the other symbols are the same as before.

It is of interest to note that this is the general expression for all three cycles. In the Diesel cycle (2) the ratio α becomes unity, and in the gas-engine cycle (1) the ratio ρ is unity. By substitution of either of these values, formula (3) becomes either formula (2) or (1), as the case may be.

[1] "Thermodynamic Cycles with Variable Specific Heat of Working Substance," by W. J. Walker, *Philosophical Magazine*, vol. xxxiv. Sept. 1917.

Variable Specific Heats and Internal Energy.—Whilst the expressions (1), (2), and (3), given in the last paragraph, form a convenient relative basis of comparison for any particular type of internal-combustion engine, no theory which does not take into account apparent variations in the value of specific heats at high temperatures can arrive at any definite forecast of the state of the working substance at all parts of the stroke.

Various theories have been put forward [1] to account for the experimental result that when a gaseous mixture is exploded in a closed vessel, the maximum temperature and pressure actually reached is only about half the calculated values if the specific heat is taken as constant. Although all these theories may have a bearing on the result, including the one which states that the specific heat of a gas increases with the temperature, for purposes of calculation it is convenient to assume values for this specific heat which would account for the whole discrepancy. The determination of such "apparent" specific heats is to a large extent bound up with the increase of internal energy of a gas as the temperature rises. It has not so far been possible, nor is it necessary, to measure the total amount of internal energy contained in any particular gas, but theory shows that this supply is increased according to the relation—

$$\text{Increase of internal energy} = \text{heat taken in} - \text{work done}$$

When a perfect gas is heated at constant volume from an absolute temperature T_0 to an absolute temperature T_1

$$\text{Increase of internal energy} = K_v(T_1 - T_0)\,^*$$

If, on the other hand, the gas is heated at constant pressure, then

$$\text{Increase of internal energy} = (K_p - R)(T_1 - T_0)$$

In these equations K_v = mean value of specific heat at constant volume
K_p = mean value of specific heat at constant pressure
R = a constant depending on the units used. Its value is $K_p - K_v$

These equations show the relation between increase of internal energy and the specific heat of any gas, provided the rise in temperature can be measured. The lower temperature (T_0) is often fixed at 100° C. (or 373° C. (abs.)), and curves are drawn which show the increase of internal energy above that temperature and which are generally referred to as internal-energy curves. The slope of such curves when plotted against temperature gives a value for the specific heat.

Before discussing this matter a little more fully it is advisable to have a clear conception of the units involved. In the first case, for practically all systems of measurement it has been customary to use the centigrade scale, and therefore the centigrade heat unit (C.H.U.). The relation between the centigrade and the Fahrenheit scale is $F = \frac{9}{5}C. + 32$, and 1 C.H.U. = 1·8 B.Th.U.; but the fact that centigrade units are being used more and more in scientific design is accustoming British engineers to think equally well in either scale.

[1] For a summary of these, see Inchley, "Theory of Heat Engines," 1920, p. 312.
* See, for instance, Inchley, "Theory of Heat Engines," 1920, p. 3.

The specific heat of a gas, either at constant volume (K_v) or at constant pressure (K_p), represents the number of heat units required to raise unit quantity of the gas through one degree. It has just been pointed out that the internal energy of a gas, or more strictly the increase in internal energy of a gas, is measured by the product of the specific heat (K_v) and the rise in temperature. Hence the units involved are the same for both specific heats and for internal energy; but as the latter is a measure of work it is often convenient to convert the heat unit into a work unit.

Quantitative values of both specific heats and the internal energy of a gas may therefore be expressed in ft.-lb. per cub. ft.

When this is measured at standard temperature and pressure it is frequently referred to as the volumetric heat of the gas.

The physical chemist records his results in gramme-calories per gramme-molecule or mol. The relation between the two systems of units is shown by Sir Alfred Ewing[1] to be 3·90 cal. per mol. = 1 ft.-lb. per cub. ft.

Occasionally internal energy is expressed in ft.-lbs. per lb., which can be done when the number of cubic feet per lb. of the particular gaseous mixture is known at the required temperature and pressure.

A third system of units in common use is C.H.U. per lb. This has the same numerical value as gramme-calories per gramme. To convert this system into ft.-lb. per lb. in ° C. multiply by Joule's equivalent 1400.

A large amount of experimental research has been carried out on the relation between the increase in internal energy of gases and their specific heats at various temperatures. This work has been collated and added to by a Committee of the British Association on Gaseous Explosions.[2] From the slope of the curve of internal energy published by this committee, H. E. Wimperis[3] deduced the following formula for the specific heat of a normal gaseous mixture at constant volume

$$K_v = 0{\cdot}152 + 0{\cdot}000075T \text{ (lb.-cal. per lb.)} \quad . \quad . \quad (a)$$

where T represents the temperature of the mixture in ° C. (abs.).

In this form the units are gramme-calories per gramme, or what is the same thing lb.-cal. per lb.

This can be converted to ft.-lb. per lb. by multiplying expression (*a*) by 1400. It then becomes

$$K_v = 212{\cdot}8 + 0{\cdot}105T. \text{ ft.-lb. per lb.} \quad . \quad . \quad . \quad (b)$$

The volumetric heat or the value in ft.-lb. per cub. ft. can be obtained from this second expression if the number of cub. ft. per lb. of the mixture at standard temperature and pressure is known. This is taken by Wimperis[4] as $\frac{1}{0{\cdot}0{,}833} = 12{\cdot}78$ cub. ft. per lb. Hence,

[1] "Thermodynamics for Engineers,' 1920, p. 238.

[2] First Report, 1908, reproduced in Sir Dugald Clerk's book on "The Gas, Petrol, and Oil Engines," 1910, vol. i. Appendix. There is a summary in Inchley's "Theory of Heat Engines," 1920, pp. 313 *et seq*.

[3] In "The Internal-Combustion Engine," 3rd ed. 1919, p. 93.

[4] *Ibid*. p. 79.

dividing (b) by 12·78 (or what is the same thing, multiply (b) by 0·07833).

$$K_v = 16{\cdot}66 + 0{\cdot}00822T \text{ ft.-lb. per cub. ft.} \quad . \quad . \quad . \quad (c)$$

Lastly, to convert it into gramme-calories per mol., divide expression (c) by 3·9 to get

$$K_v = 4{\cdot}27 + 0{\cdot}002107T \text{ cal. per mol.} \quad . \quad . \quad . \quad (d)$$

Sir Alfred Ewing [1] gives

$$K_v = 5{\cdot}2 + 0{\cdot}00086t + 0{\cdot}6 \times 10^{-6}t^2$$

in gramme-calories per mol., where t is in ° C. (not absolute).

By substituting $(T - 273)$ for t this expression becomes

$$K_v = 5{\cdot}01 + 0{\cdot}000532T + 0{\cdot}6 \times 10^{-6}T^2$$

which may be converted to

$$K_v = 249{\cdot}6 + 0{\cdot}0265T + 29{\cdot}9 \times 10^{-6}T^2 \text{ ft.-lb. per lb.}$$

since it is based on the same mixture as before (12·78 cub. ft. per lb.).

W. J. Walker,[2] from a comparison of thermo-dynamical relations with actual internal-combustion engine performance, considers that the following equation represents the value of specific heat at constant volume—

$$K_v = 220 + 0{\cdot}0623T \text{ in ft.-lb. per lb. per ° C. (abs.)}$$

and at constant pressure

$$K_p = 317 + 0{\cdot}0623T \text{ ft.-lb. per lb.}$$

The way in which Walker derived his values will be indicated in more detail in the next paragraph, but it should be noted here that he has worked with a slightly heavier average mixture, namely, one occupying 12·53 cub. ft. per lb. at N.T.P.

Fig. 90, reproduced by kind permission of the Council of the Institution of Mechanical Engineers, from the discussion on Walker's paper, shows with others the values of these specific heats plotted up to an absolute temperature of 2500° C.

The agreement between the curved and straight lines (1) and (2), derived from the B.A. results, is sufficiently close to neglect, in practical work, the T^2 term up to 2000° C., though above that value it appears to rise too rapidly to give values consistent with experimental results.

The most useful form of variable specific heat, if the B.A. results are to be used, may therefore be taken as that derived by Wimperis—

$$K_v = 16{\cdot}66 + 0{\cdot}00822T \text{ in ft.-lb. per cub. ft. per ° C. (abs.)}$$

for gas engine mixtures, which are usually measured in cub. ft.

Walker's value, on the other hand, though including gas engines, is particularly suitable for oil engines. In this case the fuel is measured per lb. Hence his units are given in ft.-lb. per lb.

[1] In his "Thermodynamics for Engineers," 1920, p. 253.

[2] "Internal-Combustion Engine Development," *Proc. Inst. Mech. Eng.* Dec. 17, 1929.

The mean value of these curves for any given temperature range, multiplied by that range of temperature in ° C., gives the increase in internal energy of the gaseous mixture in ft.-lb. per lb.

For example, taking Walker's gaseous mixture raised from 373° to 1000° C. From Fig. 90 at 1000° C., $K_v = 282$, and at 373° C., $K_v = 233$, giving a mean K_v of $\frac{282 + 243}{2} = 262.5$ for a range of temperature of $1000 - 373 = 627°$ C. Hence, increase in internal energy $= 262.5 \times 627 = 0.1645 \times 10^6$ ft.-lb. per lb.

This somewhat roundabout method is included here with the object of showing more clearly the direct connection between the specific heat of a gas and the definition of its internal energy. Reference has already been made to curves of internal energy, published by the "Gaseous Explosions" Committee of the British Association. The curve, based on Walker's values, is given in Fig. 91, on p. 201. The usual method of deriving such curves from the specific heat is to compute them direct from the relation—

$$E = \int K_v dT$$

where E = increase of internal energy between the chosen limits ot the absolute temperature (T).

Applying this to the example just given

$$\begin{aligned} E &= \int_{373}^{1000} 220 + 0.0623T dT \\ &= 220(1000 - 373) + \frac{0.0623}{2}(1000^2 - 373^2) \\ &= 138{,}000 + 26{,}700 = 0.1647 \times 10^6 \end{aligned}$$

In this connection it should be noted that the curve given in the first report of the B.A. Committee starts with a lower temperature limit of 100° C. In Sir Alfred Ewing's book,[1] this curve is redrawn for a lower limit of 0° C. Walker's curves, shown in Fig. 91, start from absolute zero (− 273° C.), and give readings throughout in ° C. absolute.

The importance of an easy means of determining the increase of internal energy during a rise in temperature lies in the fact that the total energy available at the end of explosion is, theoretically, the sum of the latent energy in the fuel plus the internal energy caused by the compression. As will be shown later, it is possible to construct an ideal indicator card with the aid of such curves for any particular conditions. A comparison of such a diagram with an actual indicator card supplies information about heat losses otherwise unaccounted for, as, for instance, heat transmission to the walls of the cylinder, or the loss of heat due to possible incomplete combustion.

In order to determine the latent energy in the fuel, it is necessary to know the net calorific value of the fuel, and to calculate the quantity of fuel present per charge. From this the calorific value per cub. ft. or lb. of cylinder contents is obtained. Multiplying this value by Joule's equivalent gives the latent energy of the fuel which is required. It is not quite a simple matter to estimate correctly the actual quantity of

[1] "Thermodynamics for Engineers," 1920, p. 252.

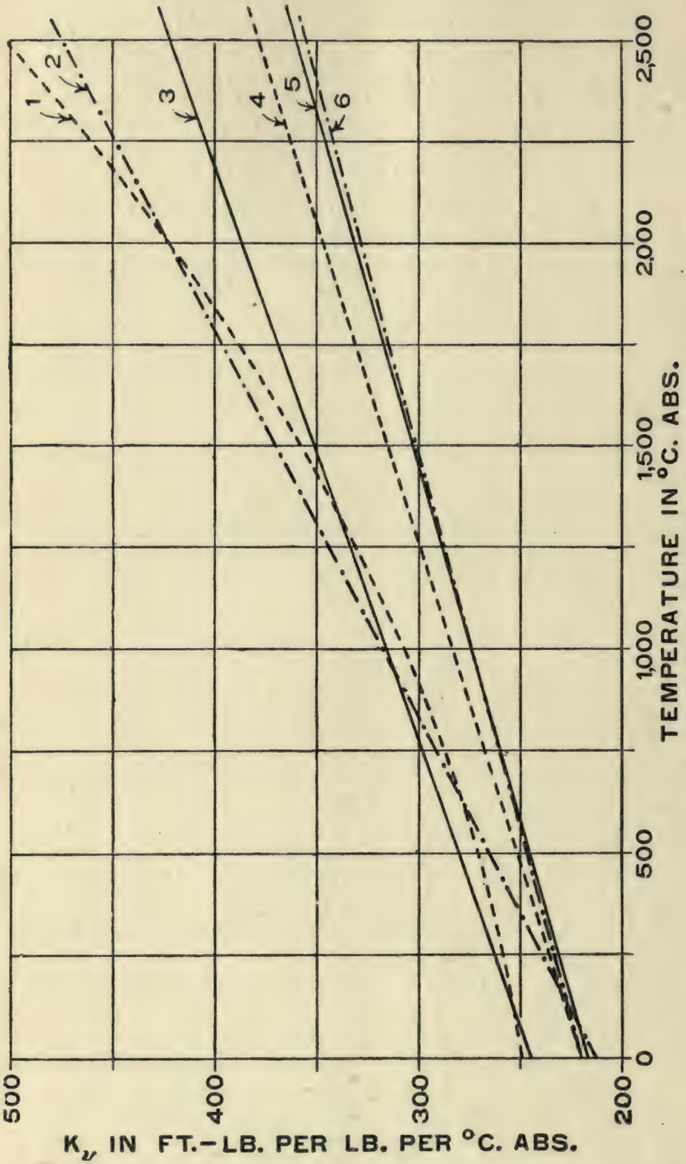


FIG. 90.—Variable specific heat curves.

[1] Curve 1.—Sir Alfred Ewing	$249{\cdot}6 + 0{\cdot}0265T + 29{\cdot}9 \times 10^{-6}T^2$
Curve 2.—H. E. Wimperis	$212{\cdot}8 + 0{\cdot}105T$
Curve 3.—Sir Dugald Clerk	$245 + 0{\cdot}0714T$
Curve 4.—W. J. Walker	$220 + 0{\cdot}0623T$
Curve 5.—Dr. Langen (for oil mixture)	$217 + 0{\cdot}0575T$
Curve 6.—Löffler and Riedler (for air)	$221 + 0{\cdot}052T$

[1] Curve 1.—"Thermodynamics for Engineers" (1920), p. 253.
Curve 2.—"The Internal-Combustion Engine" (1919), p. 79.
Curve 3.—Deduced by H. E. Wimperis, *Proc. Inst. C.E.* vol. clxix. (1907), p. 171.
Curve 4.—"Thermodynamic Cycles of Internal-Combustion Engines," *Proc. Inst. Mech. Eng.* Dec. 1920, p. 1247.
Curve 5.—Quoted in "Zeitschrift des Vereines Deutscher Ingenieure," vol. lx. (1916), p. 279.
Curve 6.—"Oelmaschinen" (1916), p. 488.

live fuel present per unit volume or weight of the cylinder contents. A method, due to the late Professor Hopkinson, is applied to a numerical example by Inchley,[1] and another method is given by W. A. Tookey for gas engines.[2] For preliminary work it is accurate enough if the ratio of air to fuel is chosen or assumed, and that proportion used without making any allowance for exhaust residue. A slightly more accurate result would be obtained if this exhaust residue is taken as 7 per cent. of the cylinder contents for low-compression engines ($r = 4\frac{1}{2} - 7$), and half that value for high-compression ratios ($r = 10$ or over).

For oil engines H. Moore[3] gives the following formula for the theoretical amount of air required to burn 1 lb. of fuel—

$$A = 0{\cdot}116(C + 3[H - \tfrac{1}{8}O]) \text{ lb.}$$

where C, H, and O are the percentages of carbon, hydrogen, and oxygen respectively in the fuel. In practice he points out that whereas petrol engines approximate closely to the theoretical quantity, heavy oil engines require from 2 to $2\frac{1}{2}$ times this amount of air when working at normal full load.

Ideal Efficiencies with Variable Specific Heats.—An expression for the ideal efficiency of the constant-volume cycle with variable specific heat was first obtained by H. E. Wimperis, and is given in his book, "The Internal-Combustion Engine."[4] S. Lees was the first to deduce an expression for the Diesel engine, or constant-pressure cycle,[5] and W. J. Walker has extended his method of analysis to the dual-combustion cycle.[6] In the same paper Walker also deduces expressions for the constant-volume cycle (which can be shown to conform with that obtained by Wimperis),[7] and for the constant-pressure cycle. For practical purposes Walker shows that three variable specific heat expressions may be derived from his formulæ by multiplying the corresponding values for constant specific heat (with a modified γ) by "diminishing factors."

The full expressions, therefore, of ideal efficiencies allowing for variable specific heats may, in the light of present knowledge, be taken as follows:—

(1) Constant-volume combustion

$$1 - \left(\frac{1}{r}\right)^{\gamma_0 - 1} \left[1 - \frac{\lambda T}{2}(a + 1)\right]$$

(2) Constant-pressure combustion (Diesel type)

$$1 - \left(\frac{1}{r}\right)^{\gamma_0 - 1} \times \frac{\rho^{\gamma_0} - 1}{\gamma_0(\rho - 1)} \left[1 - \frac{\lambda T}{2}(\rho + 1)\right]$$

[1] Inchley, "Theory of Heat Engines," 1920 ed. pp. 324–327.

[2] Tookey, "Internal-Combustion Engine Tests," *Proc. Inst. Mech. Eng.* 1914, Parts 1 and 2, p. 5.

[3] Moore, "Liquid Fuels for Internal-Combustion Engines," 1918, p. 174.

[4] Wimperis, "The Internal-Combustion Engine," 3rd ed. 1919, p. 86. The way in which this expression is derived is shown on p. 81 of the 1908 edition of this book.

[5] *Engineering*, vol. xcix. (1915), p. 1.

[6] "Thermodynamic Cycles with Variable Specific Heat of Working Substances," by W. J. Walker, *Phil. Mag.* vol. xxxiv. Sept. 1917.

[7] *Proc. Inst. Mech. Eng.* 1920, p. 1304.

(3) Dual-combustion

$$1 - \left(\frac{1}{r}\right)^{\gamma_0 - 1} \times \frac{a\rho^{\gamma_0} - 1}{a - 1 + \gamma_0 a(\rho - 1)} \left[1 - \frac{\lambda T}{2}(a + \rho)\right]$$

In each of these three cases the "diminishing factor" is represented by the term within square brackets.

The symbols have the following meanings:—

$$r = \frac{\text{cylinder volume before compression}}{\text{cylinder volume after compression}}$$

$$\rho = \frac{\text{cylinder volume at "cut off"}}{\text{cylinder volume after compression}}$$

$$a = \frac{\text{maximum pressure after "firing"}}{\text{pressure at end of compression before "firing"}}$$

T = suction temperature in ° C. (abs.) (often taken at 373° C. (abs.)).

If the formulæ for the specific heats are written in the form

$$K_p = A + ST \quad \text{and} \quad K_v = B + ST$$

where A, B, and S are constants, then

$$\gamma_0 = \frac{A}{B} \text{ (where ordinarily } \gamma = \frac{K_p}{K_v} \text{ at N.T.P.)}$$

$$= \frac{317}{220} \quad \text{or } 1{\cdot}442\text{, from Walker's values on p. 196}$$

$$\lambda = \frac{S}{B} = \frac{0{\cdot}0623}{220} \quad \text{or } 0{\cdot}000283\text{, from the same source}$$

It was a consideration of the value of λ when inserted in the above expressions which helped Walker to obtain his numerical constants A, B, and S. If the value is taken much higher it makes it possible to obtain efficiency ratios of more than 100 per cent. from engines in actual practice.[1]

The form of these expressions show very clearly a point which has already been mentioned, namely, that for comparative purposes the diminishing factor may be neglected in any given type of combustion, but that when the efficiency ratio of any particular engine is required, to ascertain how nearly the possible ideal is being approached, it is essential to make allowances for the variable specific heat factor.

The Temperatures of the Working Substance.—If an actual indicator card is available, and the temperature in degrees absolute at one point (1) known or estimated, the temperature at any other point (2) can be calculated from the relation

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

provided there has been no chemical contraction due to combustion.

If the points lie on either side of the ignition, and the products of combustion are x times the specific volume of the original working substance, then

[1] See *Proc. Inst. Mech. Eng.* 1920, p. 1306.

$$\frac{xP_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

The value of x may be calculated from an analysis of the exhaust gases,[1] but for an approximate estimate it may be taken as 0·97 for a gas engine and 0·99 to 1·00 for a Diesel oil engine. The ordinates of the card can be drawn directly on the card, if the correct compression ratio is known. The zero-pressure line is, of course, 14·7 lb. below the atmospheric line drawn to the scale of the spring used. The zero-volume line is drawn at a distance $r - 1$ from the "toe" of the diagram.

If the clearance volume or the compression ratio is not known, it can be calculated by an approximate method given in Inchley,[2] or by a method shown in a paper by W. A. Tookey,[3] which gives the use of a light spring card to deduce the "volumetric efficiency," or the ratio of the effective piston displacement to the specific piston displacement. A method of deducing both the correct compression ratio and the temperature at any part of the stroke without calculation is given in the next paragraph.

Walker's Chart for Internal-Combustion Engine Calculations.—Walker has embodied the results of his investigations into the variable specific heats of gaseous mixtures in a chart which is reproduced here by kind permission of the Institution of Mechanical Engineers.[4]

The chart is based on the special though fairly general case of 1 lb. of working fluid occupying 18 cub. ft. at 373° C. (abs.), and 14·0 lb. per sq. in. pressure. The starting point under these conditions is the end of the suction stroke, or what is the same thing, the beginning of compression. The lowest hyperbolic type of curve running from the bottom right-hand corner to the top left hand of the chart is the compression curve of this substance drawn to pressure in lb. per sq. in. (left-hand scale), and volume in cub. ft. (bottom scale). The remaining six curves of this type, read to the same scales, are expansion curves, which are used to determine hypothetical indicator cards for any given cycle or conditions. All these curves have been obtained from the relation for adiabatic changes under variable specific heat, namely

$$PV^{\gamma_0}e^{\lambda\frac{PV}{R}} = \text{constant}^{5}$$

where $\gamma_0 = \frac{A}{B} = 1·44$

$\lambda = \frac{S}{B} = 0·000283$

e = base of Napierian logs = 2·7183

$R = A - B = 97$

[1] Inchley, "Theory of Heat Engines," 1920 ed. p. 236. See also p. 326 for an application of this method.

[2] Inchley, "Theory of Heat Engines," 1920 ed. p. 294.

[3] W. A. Tookey, "Internal-Combustion Engine Tests," *Proc. Inst. Mech. Eng.* Jan. 1914, p. 9.

[4] W. J. Walker, "Internal-Combustion Engine Development," *Proc. Inst. Mech. Eng.* Dec. 1920, p. 1255.

[5] See Inchley, "Theory of Heat Engines," 1920, ed. p. 323.

The straight (volume) lines radiating from the origin are to be read in conjunction with the pressure scale on the left and the absolute temperature scale on the top of the diagram. These are obtained from the relation

$$\frac{PV}{T} = \text{a constant}$$

the contraction coefficient (x) being neglected.

They enable the temperature to be read off direct when the pressure is known.

Lastly, the two curves labelled constant-pressure energy line and internal (or intrinsic) energy line are read from the temperature scale at the top and the energy scale at the right-hand side of the diagram. These curves have been drawn from the relation already mentioned (p. 197)—

$$E = \int K dT$$

The chart may be used in two main ways—

(*a*) Given or assuming the correct compression ratio and the lower calorific value of the fuel, an indicator card may be drawn from which the probable mean effective pressure can be obtained in the usual way, and hence the dimensions of the engine cylinder. The use of the chart in this connection will be explained later on when the question of design is considered (p. 247).

(*b*) Given an indicator card of any kind of internal-combustion engine, the compression ratio and temperature at any point of the stroke may be obtained.

To deduce the compression ratio read off the maximum compression pressure from the card, and note from the compression curve of the chart what volume this corresponds to. The compression ratio will then be 18 divided by this volume.

Re-draw the indicator card on tracing paper, so that the pressure scale corresponds with that of the chart, and the volume scale is such that the card lies between the compression volume just obtained and 18 cub. ft. The temperature can then be obtained by striking horizontally from any required pressure point to the radiating volume line, which corresponds to the volume of that point as read off the bottom scale, and then reading the temperature on the scale at the top of the chart. Such readings can only be obtained with a sufficient degree of accuracy if the chart is reproduced on a larger scale, say the size of an imperial sheet of drawing paper.[1]

[1] Reference should also be made to an article by T. B. Morley in *Engineering*, vol. cxii. (1921), p. 302, which gives, amongst other useful information, a nomographic chart, embodying Walker's values, for finding P, V, T, and E.

CHAPTER XI

DESCRIPTIONS OF INTERNAL-COMBUSTION ENGINES

Gas Engine Types.—Gas engines have now been in successful commercial use for forty years, and their design is already beginning to show the effect of inherited tradition so noticeable in the steam engine. Practically all gas engines operate on the constant-volume cycle, which means that the air standard of efficiency is applicable for comparative purposes, and the modified air-standard formula on p. 199, for determining how near any particular engine is approaching the possible ideal. They may, however, be grouped into two main classes, depending on whether this cycle is carried out in four strokes or in two strokes of the piston. The main types which have been developed in large sizes fall roughly under the following headings. The addition of maker's names to this list is included for the sake of identification, as it has become a common practice in internal-combustion engineering to use such names to designate a particular type. On the other hand, it should be noted that there are many other excellent firms building similar types, and that conversely the firms mentioned do not necessarily confine themselves to the type in question.

Four-stroke Gas Engines (Otto cycle).

(*a*) Single-cylinder, single-acting, horizontal (Crossley).
(*b*) Tandem, double-acting, horizontal (Nuremberg).
(*c*) Tandem, single-acting, vertical (National).
(*d*) Single-cylinder, double-acting, vertical (British Westinghouse).[1]
(*e*) Pistonless, single-acting, vertical (Humphrey pump).

Two-stroke Gas Engines (Clerk cycle).

(*f*) Single-cylinder, single-acting, horizontal, with two opposed pistons (Oechelhauser).
(*g*) Single-cylinder (or tandem), double-acting, horizontal (or vertical) (Koerting).
(*h*) Solid-piston Humphrey pump.

(*a*) **The Crossley Gas Engine.**—By far the largest number of gas engines in existence are of the single-cylinder, single-acting, horizontal type working on the four-stroke cycle. This type has proved to be particularly reliable for small gas engines ranging from $\frac{3}{4}$ H.P. to 300 H.P. per cylinder, and larger sizes are made by building in more than one cylinder on a single bedplate. Such cylinders may be

[1] Now the Metropolitan-Vickers Electrical Co., Ltd.

arranged either side by side or opposed to one another, or two cylinders may be worked tandem on one piston rod.

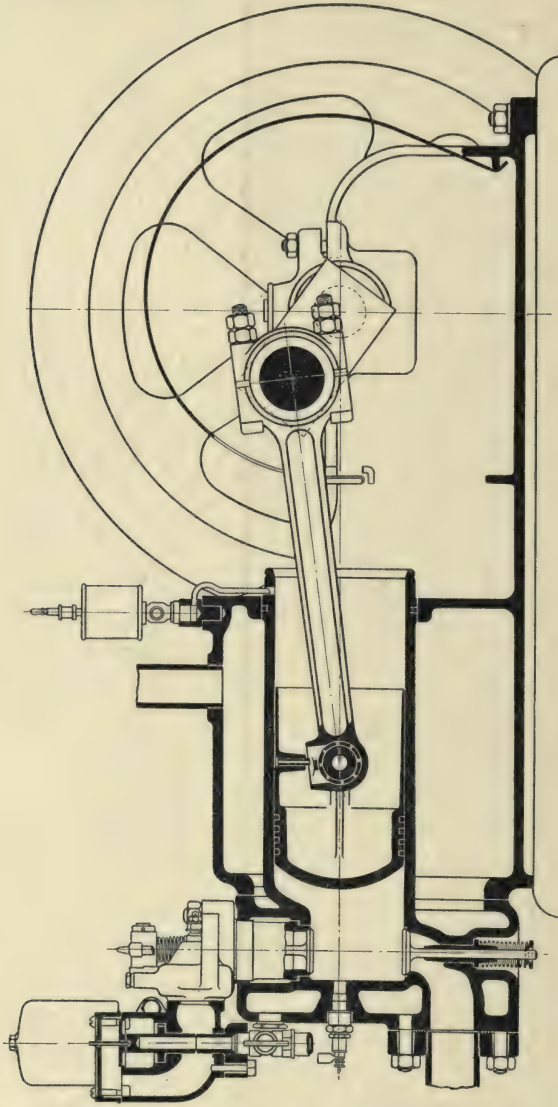


FIG. 92.—6 H.P. Crossley gas engine.

A 6 H.P. engine of this type, as made by Messrs. Crossley Brothers, Ltd., is shown in Fig. 92.[1]

The gas enters through a cock and valve shown in the top left-hand corner, where it is mixed with its right proportion of air. The mixture

[1] From *The Engineer*.

then passes through a throttle valve, which is not shown in section, but which consists of a number of ports which can be covered or uncovered by a small balanced piston. The position of this piston, and therefore the quantity of mixture passing to the inlet valve, is controlled by a centrifugal governer (not shown), which is driven directly from the crankshaft by a worm-and-wheel shaft. The inlet valve (above) and the exhaust valve (shown below) are cam operated in these small sizes by a second shaft driven from the same worm at half the speed of the engine. The combustion chamber is so designed that any deposit which may collect is swept out through the exhaust. The mixture is fired by a sparking plug shown in the back end of this combustion chamber, which is operated by a high-tension magneto driven from the same shaft as the governor. The breech end and cylinder liner are in one piece in these small engines, and the ample water jacket surrounding both the cylinder and the breech end should be noted. There is no crankshaft balance-weight, but two flywheels are provided, one on each side. The engine is particularly suitable for agricultural requirements, or for driving a small workshop without attention. In the larger sizes, Fig. 93, the cylinder liner is made in a separate piece, the crankshaft is balanced, and only one flywheel is fitted. In all sizes the girder type of bedplate is now used, which extends underneath the full length of the cylinder proper. Such engines are made up to about 120 B.H.P. per cylinder when running at 180 R.P.M., but are also built for double that power when two cylinders are placed side by side. Figs. 92 and 93 are reproduced by the courtesy of the firm of Crossley Bros., Ltd., Openshaw, Manchester.

(*b*) **The Galloway Gas Engine.**—Engines of the tandem, double-acting, horizontal type are often referred to as "Nuremberg" engines, from the fact that the Maschinenbau-Gesselschaft Augsburg-Nurnberg (M.A.N.) were one of the first firms to develop it. The type, slightly modified in mechanical construction, is built in England by Messrs. Galloways, Ltd., of Manchester.

By the kind permission of this firm a cross-sectional drawing of their engine is shown in Fig. 94, and a detail of one of the cylinders in Fig. 95. There are two similar cylinders placed tandem on one piston rod, and each cylinder has two separate sets of inlet and exhaust valves. Each of these sets, working on the four-stroke cycle, provides one working stroke in every two revolutions. The four sets of valves are so timed that one impulse occurs for each stroke, giving a total of two single-cylinder impulses per revolution. The gas and air enters the inlet valve along the two passages shown at the top left-hand of the end elevation of the cylinder. These passages are controlled by wing throttle valves, as shown. The gas passage, which is above, has two throttles, the one furthest away from the valve being hand operated for control purposes, and the other gas throttle is connected by links to the air throttles and arranged so that at light loads the gas is more throttled than the air. The spindle of this second gas throttle extends through the valves of both cylinders, and is controlled by a servo-motor governor driven off the cam shaft. If wide variation of speed is required, as for instance when used with blowers, these valves can be arranged for hand control with an emergency governor. Further control is supplied

by means of the ignition gear, and gas stop valves and wing air valves for each cylinder.

The inlet and exhaust valves, as can be seen from Fig. 95, are of the mushroom or drop-valve type, and are operated from a single cam

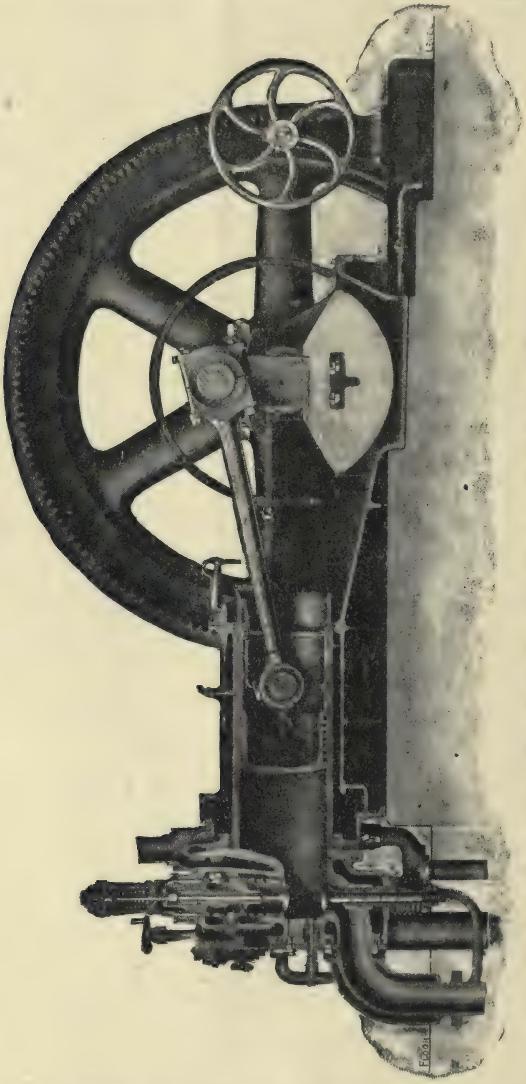


FIG. 93.—Single-cylinder, single-acting, horizontal gas engine (Crossley).

shaft by means of rocking levers and carefully designed springs. The modern type of inlet valve is double, the upper or gas valve being mounted flexibly on the valve spindle, so that both valves may close tightly even if the seatings wear unevenly.

There are three ignition plugs in each cylinder end, as can be seen in the end elevation of Fig. 95, and low-tension ignition at about 70 volts is usually installed by this firm.

The water-cooling arrangements can be clearly seen in the drawings. The cylinder is water-jacketed, and also the exhaust valve box and valve seat. In some makes the exhaust valve itself is also water-cooled. The piston rods are hollow and water under pressure is supplied through a rocking pipe, seen between the two cylinders in Fig. 94, to the coupling between the two rods, and thence to each piston and back again to the rocking pipe.

The constructional details embody the result of considerable experience, the separate cylinder liner, the cylinder cast in two halves and the barrel casing of the jacket added afterwards, the water-cooled cylinder ends, the adjustable piston-rod nuts, and the design of the

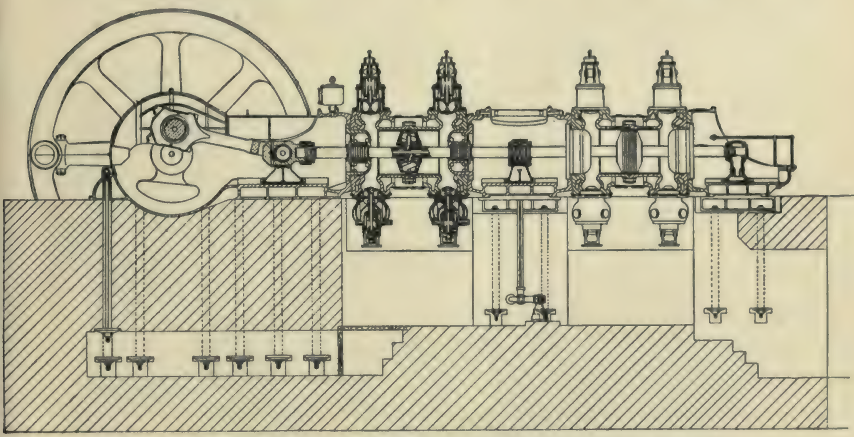


FIG. 94.—Tandem, double-acting, horizontal gas engine (Galloway).

piston itself should be noted. Many of these points are controlled by patents.

Provision is made to take up longitudinal expansion by arranging for the distance piece between the cylinders and the tail-rod support to be free to slide on sub bedplates.

Engines of this type are made in sizes varying from about 500 to 3000 B.H.P. at normal full load. They are capable of 10 per cent. over load for several hours at a stretch, and can give 20 per cent. above normal rating for short periods. The size illustrated has cylinders 45·3 in. diameter by 51·2 in. stroke, and gives 1850 B.H.P. when running at 94 R.P.M. As these engines are usually designed to run on waste gas from blast furnaces or coke ovens their consumption is not of primary importance. They average about 10,000 B.Th.U. per B.H.P. per hour, and have a brake thermal efficiency round about 30 per cent. The compression ratio is low, $4\frac{1}{2}$ to 5, and the mean effective pressure in the Galloway engine 60 lb. per sq. in., corresponding to a maximum explosion pressure of about 300 lb. per. sq. in. Continental practice

is often higher, about 75 lb. per sq. in. M.E.P., and 350 lb. per sq. in. maximum pressure. These must be taken as average figures, as considerable variation exists in individual cases.

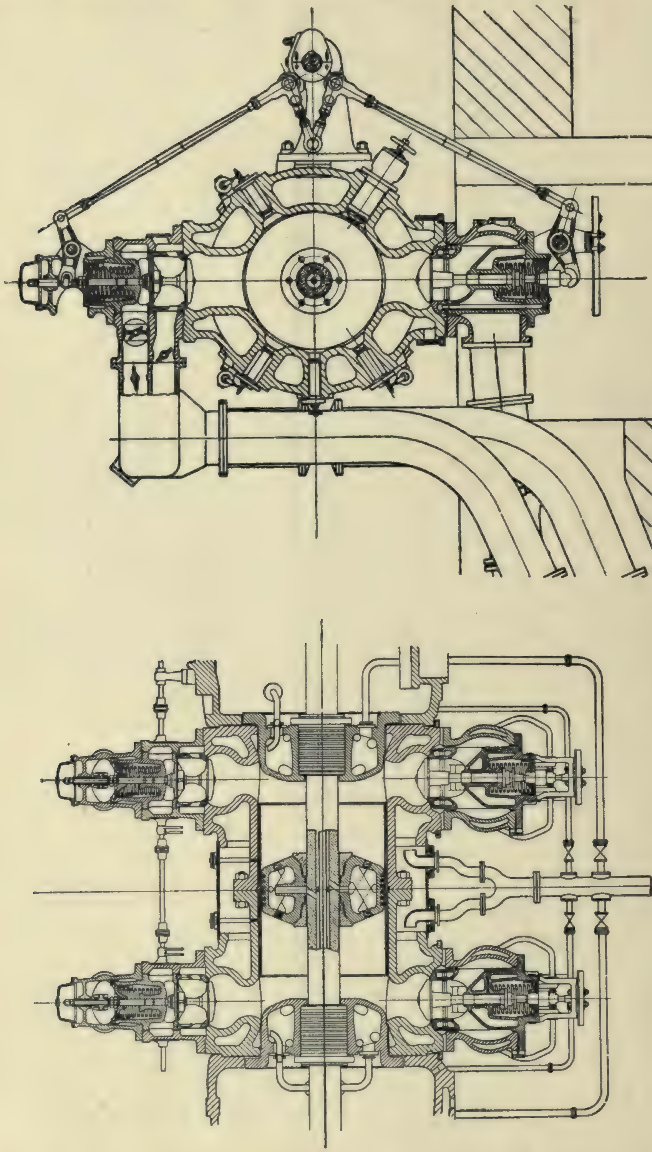


FIG. 95.—Cylinders of Galloway gas engine.

On the continent an engine of similar design to Galloway's is built by Erhardt & Sehmer. One of the oldest designs of the Nuremberg

type was made by the Société Cockerill, of Belgium,[1] and the type has been developed in America by such firms as the Snow Steam Pump Works, of Buffalo, N.Y.

(*c*) **The National Gas Engine.**—An example of a modern tandem, single-acting, vertical gas engine working on the four-stroke cycle is shown in Fig. 96 by the courtesy of the makers, the National Gas Engine Co., Ltd., of Ashton-under-Lyne, near Manchester. The illustration shows the smallest standard size made by this firm, and develops 300 B.H.P. when running at 300 R.P.M. The stroke is 18 in. Engines of similar design are made up to 1500 B.H.P. when running at 200 R.P.M. In this case the stroke is increased to 24 in., and there are 6 cranks.[2]

The gas and air pipes are seen at the top right hand of the side elevation. They pass through two throttle valves, actuated by a centrifugal governer on the end of the crankshaft, but are kept separate until they reach the inlet valve shown in section at the top of the end elevation. In the larger sizes this inlet valve has two valve faces and seatings. The cylinders are single-acting, and by firing alternatively give one working stroke per crank for every revolution. The valves are operated by rocking levers and rods actuated by cams on the half-time shaft. The spur-gear drive of this shaft from the crankshaft can be seen next to the governor.

A feature of this engine is the intermediate cylinder head, which is made to form a buffer cylinder underneath the top piston. The air in this space is alternatively compressed and re-expanded as the pistons move down and up. This cushioning effect makes for smooth running and improves the evenness of the twisting moment. No moving parts are water-cooled in this engine, but the cylinders and valve chests are surrounded by water jackets and the intermediate cylinder head is water cooled. Forced lubrication is used throughout. The valveless oil pump for this purpose can be seen in section at the bottom of the end elevation. It is driven by means of an eccentric from the crankshaft. High-tension ignition is used, and there are two independent sparking plugs to each cylinder. These are placed either in the valve chest or in the passage between the valves and the cylinder head. The magnetos for operating these plugs can be seen driven off the right-hand end of the cam shaft. The engine, like most large gas engines, is started by air compressed to between 250 and 350 lb. per sq. in. This air is passed through timing valves worked by the cam shaft to automatic inlet valves shown at the top right hand of the lower cylinder. These engines will run on waste gases or producer gas. A mixing arrangement can be fitted, which allows the engine to be run on a rich gas like coal gas without adjustment. The coal gas is mixed with exhaust gases or producer gas as it passes through the throttle valve of the governor. The effect is to reduce the heat value to one for which the engines are designed, and the change over from one gas to another can be made without stopping the engine.

It should be noted that the upper cylinders are about 1 in. larger

[1] See p. 246.

[2] For description and drawings of this largest size, see *Engineer*, vol. cxxvi. (1918), p. 207.

in the bore than the lower ones. This greatly facilitates assembly and the removal of the reciprocating parts. The piston rod is a hollow cast-iron sleeve with a forged steel rod passed through it to lock the two pistons together.

An engine of this type but with opposed cylinders has been made

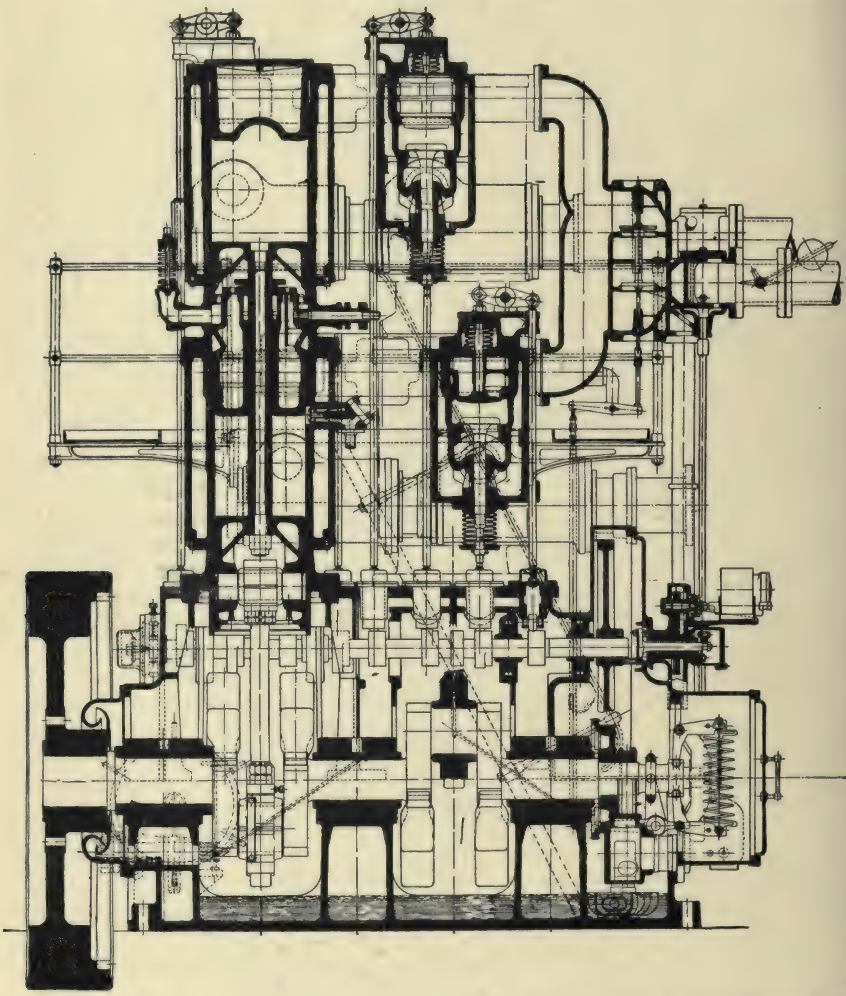
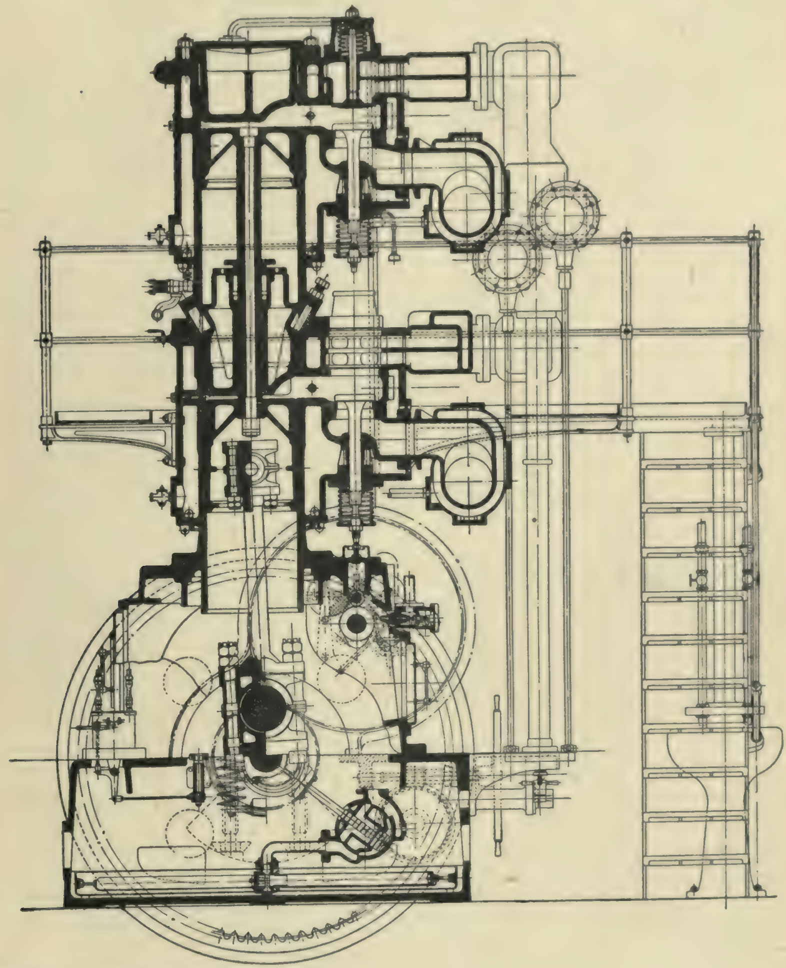


FIG. 96.—Tandem single-acting vertical

by the Premier Gas Engine Co., of Nottingham, which has distinctive features of mechanical design, including a triangular connecting rod which transmits the motion of two sets of pistons to a single crank. Full details with sectional drawings and photographs of this engine can be seen in the *Engineer*, vol. cxxix. (1920), p. 526.

(*d*) **Single-cylinder, Double-acting Vertical Type.**—The Metropolitan-Vickers Electrical Co., of Manchester, at the time when they were known as the British Westinghouse Co., have also built vertical four-stroke cycle gas engines, with three double-acting cylinders working on three cranks. A description with



gas engine (National).

photograph, of one of the largest of this type can be found in the *Engineer*."[1] This engine had cylinders 26 in. diameter by 30 in. stroke, and was rated at 1080 B.H.P. when running at 166½ R.P.M. on producer gas with a lower calorific value of 135 B.Th.U.

[1] *Engineer*, vol. cxxv. (1918), p. 452.

(*e*) **The Humphrey Gas Pump**—In November 1909 H. A. Humphrey read a paper before the Institution of Mechanical Engineers, describing the application of the principle of internal combustion to the direct pumping of water. Various types of pumps were shown both for four-stroke and two-stroke cycle operation. All Humphrey pumps are made without a fly-wheel in which the cycle is controlled by the inertia of oscillating masses and by the elastic properties of gas cushions. The method was first embodied in four large and one smaller pumps used in connection with the Chingford reservoir waterworks of the Metropolitan Water Board. A diagrammatic cross-section through one of these pumps can be seen in Fig. 97, reproduced from the *Engineer*,[1] by kind permission of the Editor. In its essentials it will be noted that the pump consists of a long "play" pipe with two right-angle vertical bends. One end

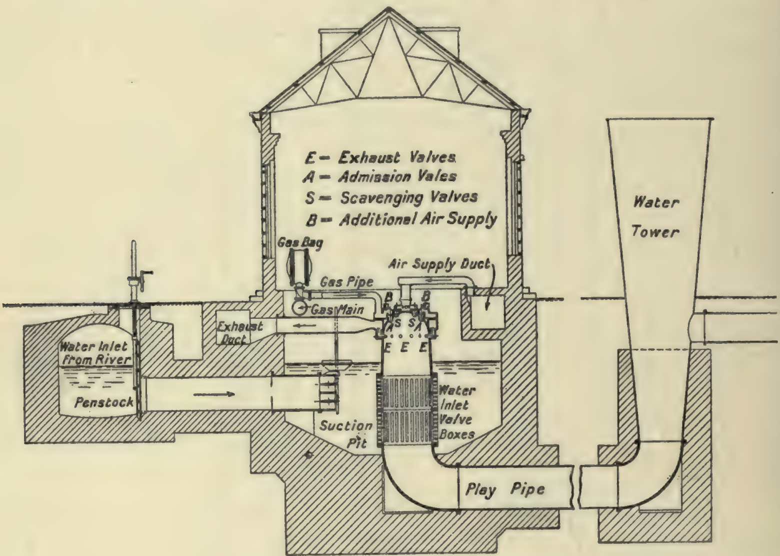


FIG. 97.—Humphrey pump at Chingford.

terminates in a "water tower" open to the atmosphere and has the delivery pipe branching from it, the other end is closed to form the combustion chamber for the working substance. This pipe is filled with water up to the level shown in the power house, and air and gas are introduced above by a compressor and exploded by electric ignition on the face of this water. Round the top of the combustion chamber are arranged, in a vertical position, eight air-scavenging valves and eight gas inlet valves, each with an additional air-supply valve. These valves work automatically against springs, but an interlocking arrangement worked by the movement of the water prevents both sets opening

[1] *Engineer*, vol. cxv. (1913), p. 272. This volume contains a very full description of the waterworks as well as detailed sectional drawings of the actual pumps. These were made by the firm of Siemens Bros., to the design of the Pump and Power Co., Ltd.

together. Round the sides horizontally are placed sixteen exhaust valves. These valves are normally held open by springs, and have non-return valves behind them to prevent admission through the exhaust. They are also connected to the interlocking device which holds them closed during the suction stroke. The water-inlet valves are also automatic, and consist of a series of long slots round the base of the combustion chamber closed by metal flaps held up by springs.

When the working substance is exploded, the whole column of water is set in motion and forced down past the inlet valves and up in the water tower. Some of it is thus passed along the delivery pipe. The impetus carries the pressure inside the combustion chamber below atmospheric pressure and opens the automatic air-scavenging valves, which has the effect of diluting the burnt gases. The difference in level between the water inside and outside the combustion chamber causes the water-inlet valves to open. The incoming water, together with the return of the swinging column, closes the air scavenge and forces the diluted products of combustion through the exhaust valves, until these are reached and closed by the water itself. As all valves are now closed the swinging water column is gradually brought to rest by the cushioning effect of the remainder of the gases trapped in the head of the combustion chamber. The water column swings back, but this time does not travel so far, giving a shorter stroke to draw in fresh air and gas through the admission valves, whilst the exhaust valves are held closed. The column returns a second time on its second and shortest stroke, and compresses the working mixture ready for ignition as the cycle repeats itself. There are about nine impulses per minute. The four large pumps have 6 ft. diameter play pipes, 48 ft. long between bends, and 7 ft. diameter combustion chambers; the smaller pump has a 4 ft. 3 in. diameter play pipe and 5 ft. diameter combustion chamber. The gas is obtained from Dowson producers using anthracite, and the following table gives the summary of results of tests made on all five pumps[1]:—

Pump No.	1	2	3	4	5
Number of tests	6	6	6	6	6
Average duration of tests in minutes	9·27	8·95	8·37	9·67	10·0
Lift in feet	30·01	30·24	30·06	32·6	30·24
Water pumped, galls. per minute	33,407	32,773	33,047	32,663	18,116
P.H.P. developed	303·9	300·4	301·1	322·7	166·0
Gas used at N.T.B. in cub. ft. per min.	395·4	393·3	391·5	400·1	191·6
Lower cal. value of gas B.Th.U. per cub. ft.	145·7	146·4	146·2	142·2	138·1
Average thermal efficiency per cent.	22·39	22·19	22·33	24·07	26·63
Anthracite used per P.H.P. per hour in lb.	0·946	0·957	0·949	0·881	0·796

[1] H. de P. Birkett, "The Humphrey Pumps at Chingford," Woolwich Polytechnic Engineering Society, Nov. 6, 1913.

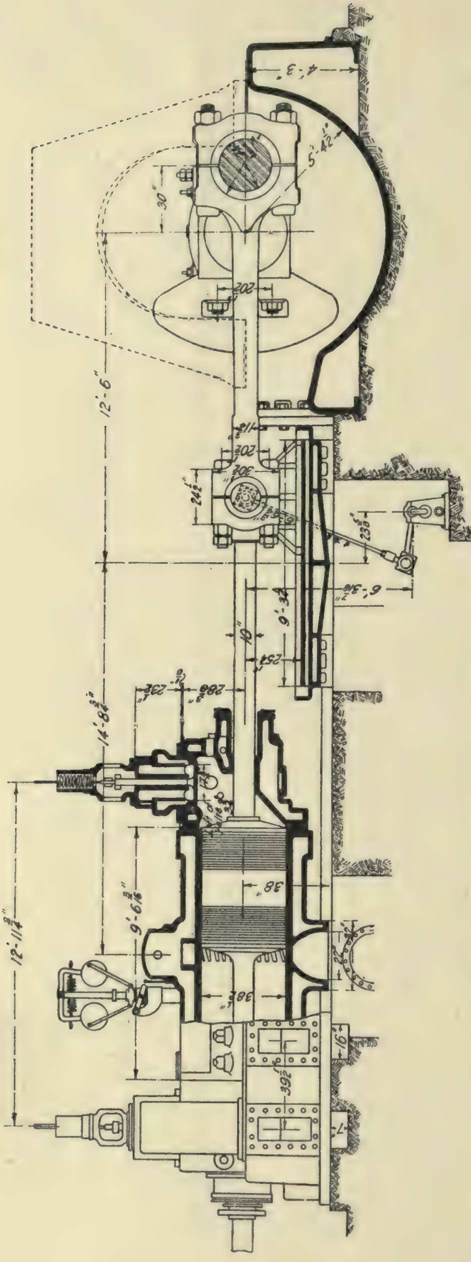


FIG. 98.—Two-stroke, double-acting, horizontal gas engine (Koerting type).

Following on the successful working of these pumps, an order was placed by the Egyptian Government for a large pumping plant to be erected near Alexandria, and each Humphrey pump unit was to deliver one hundred million gallons per day with a lift of 19 feet. The whole installation was designed for ten units, but the contract was cancelled for financial reasons soon after the outbreak of war.

Two-stroke Gas Engines.

(*f*) **Single-cylinder, Single-acting, Horizontal, with two opposed pistons (Oechelhauser Type).**—This type of engine is now practically obsolete as far as large gas engines are concerned, but it is interesting as the parent of the Junkers Diesel engine, and in a modified form of the Fullagar oil engine (p. 236). There are two opposed

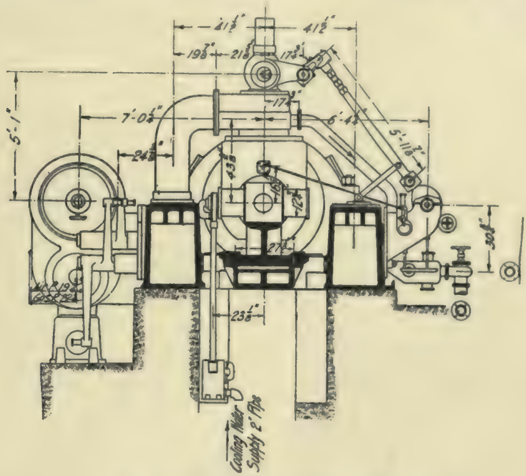


FIG. 98A.—End elevation of Fig. 98.

pistons in a trunk cylinder working on a single crankshaft, to which the outer piston is connected by a cross beam and two connecting rods down each side of the cylinder. For sectional drawings and further details, reference can be made to a paper by R. E. L. Chorlton, read before the Manchester Association of Engineers in 1911.[1]

(*g*) **Single-cylinder, Double-acting, Horizontal Two-stroke Gas Engine (Koerting type).**—An interesting example of the Koerting type of gas engine is shown in Fig. 98, through the courtesy of the American Society of Mechanical Engineers.[2] It represents one side of a twin-cylinder engine made by the De La Vergne Machinery

[1] "Large Gas Engines of the Two-cycle Type," *Trans. Manchester Association of Engineers*, 1910–11, pp. 369 *et seq.* Reprinted in *Engineering*, vol. xci. (1911), p. 325.

[2] Paper by E. P. Coleman on "The First Large Gas-engine Installation in American Steel Works," *Trans. Am. Soc. Mech. Eng.* vol. xxxii. (1910), p. 1384.

Co., New York, to the designs of the firm of Koerting Brothers, Hanover. The engine forms one of 16 similar units installed for blowing purposes in the Buffalo works of the Lackawana Steel Co. This engine works on the two-stroke cycle and is double acting, so that there are two impulses per cylinder per revolution, or four impulses per revolution with twin cylinders working on a single crankshaft. It resembles in design the uniflow steam engine (see p. 122), in that the working substance is admitted at each end of a long cylinder and the exhaust takes place through a ring of central ports uncovered by the piston itself. Gas and air are compressed to about 10 lb. per sq. in. above atmosphere in two separate pumps seen on the left of the end elevation. Each of these pumps is connected to the mixing chamber above each of the two inlet valves, the gas coming to the outer ring and the air to the central part.

The timing of admission to this mixing chamber is controlled by eccentric-driven piston-delivery valves on each pump. When the inlet valve to the cylinder first opens air alone is being delivered, which sweeps the exhaust gases out through the central parts of the cylinder. The gas is then admitted and forms a working mixture, which is compressed as soon as the piston covers the exhaust ports on the return stroke, and the inlet valve is closed. Governing is obtained by means of a bye-pass valve on the gas-pump suctions, which regulate the amount of gas allowed to go forward and so vary the strength of the working mixture. Ignition is by electric spark, two plugs being provided for each cylinder head.

Each cylinder is $38\frac{1}{2}$ in. diameter by 60 in. stroke, and develops 1000 I.H.P. when running at about 60 R.P.M., and there are two cylinders working side by side on one crankshaft to form one unit.

The engines, working on blast-furnace gas, are used to drive blowers, and the paper contains detailed quantitative tests for delivering blast pressures varying from 7·65 to 27·5 lb. per sq. in.

With an air blast of 23 lb., for instance (test No. 32), the engine developed gross 2110 I.H.P. at 61·65 R.P.M., from this must be deducted 205 I.H.P. for the air pumps, 290 I.H.P. for the gas pumps, and 282 I.H.P. for friction, leaving a net B.H.P. of 1333, or a mechanical efficiency of 63 per cent. Indicator cards of this test show that the compression pressure was about 106 lb. per sq. in. and the maximum explosion pressure 175 lb. per sq. in. above atmosphere. The average mean effective pressure was $52\frac{1}{4}$ lb. per sq. in. The consumption was 14,343 B.Th.U. per B.H.P. per hour, using blast-furnace gas with a (lower) calorific value of 92·8 B.Th.U. per cub. ft. under standard conditions.

A development of this type can be seen in the *Engineer*, vol. cxxii. (1916), p. 437, where a description is given with sectional drawings and a photograph of a vertical four-cylinder double-acting two-stroke engine that has been made by the firm of Mather & Platt, of Manchester. This engine was direct-coupled to a 550 K.W. alternator, and ran at 250 R.P.M.

(*h*) **The Solid Piston Two-stroke Humphrey Pump.**—The novel principle upon which the pistonless Humphrey gas pump works relies on a swinging mass of water in a play pipe, and for small sizes

the necessary length of such a pipe is not always a convenience. To obviate this Humphrey has designed a type of small pump, which works on the two-stroke cycle, in which is embodied a moving solid mass of metal rigidly attached to an ordinary internal-combustion engine piston. A description and diagram (Fig. 99) of this type of pump are reproduced here by the courtsey of the editor of the *Engineer*.[1]

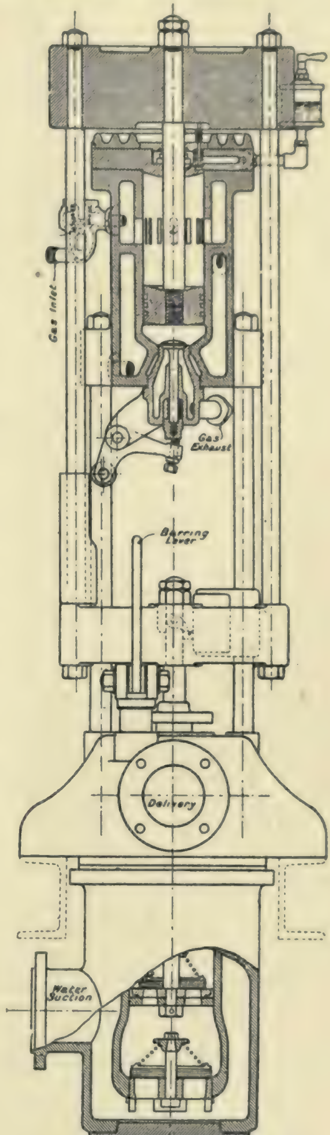


FIG. 99.—Solid-piston Humphrey pump.

The piston, seen above, is rigidly connected to an upper crosshead, which consists of a circular mass of solid metal and which in turn is connected by two rods to a lower crosshead to which the pump rod is fixed. This pump rod carries a bucket with a non-return flap valve, which works in a pump barrel closed at the bottom end by a somewhat similar foot valve.

The internal-combustion cylinder is surrounded by a jacket divided into two portions. The lower portion forms a water jacket, and into the upper portion the mixture of gas and air are admitted through the automatic valve shown on the left. A ring of slotted ports connect this mixing chamber with the cylinder. At the bottom end of the cylinder an exhaust valve is placed, which is operated by a bell-crank lever working on a cam block fixed to one of the crosshead tie rods.

The action is as follows. The piston is lifted to the top of the stroke by means of a barring lever, or ratchet arrangement, shown underneath the lower crosshead to the left of the pump rod. The piston overruns the inlet ports and a gaseous mixture is allowed to accumulate underneath. The lever is then tripped and the piston falls under gravity, compressing the charge underneath it to about 200 lb. per sq. in. and drawing in a fresh charge above. The mixture is fired

[1] *Engineer*, vol. cxxi. (1916), p. 362.

by electric ignition (not shown) and the pump starts away. On the upward stroke the fresh charge is compressed in the mixing chamber until the piston overrides the inlet ports. A small amount trapped above the piston has a cushioning effect, which brings the moving parts smoothly to rest. The exhaust valve opens on the up stroke before the inlet ports are uncovered, and the bulk of the products of combustion escape before the incoming fresh charge sweeps the remainder away. The exhaust valve closes at the same point on the downward stroke.

It will be noticed that Humphrey with characteristic boldness has inverted the usual arrangement for two-stroke cycles. He admits the fresh charge in the middle and exhausts at the end of the cylinder. All the suction and delivery of the pump are carried out on the up stroke.

On the downward stroke, water, which had been drawn into the suction chamber on the previous upward stroke, is merely transferred to the other side of the pump bucket as it sinks under gravity with an open valve. A rubber buffer, or in the larger sizes an air cushion, is placed on the top of the engine cylinder, in case the cushioning effect of compression should fail. The pump parts rest on this when not in use.

It will be noted that there are no rotating parts whatever, the only bush being that of the exhaust-valve rocking lever. All the stress is applied vertically upwards and there is no side thrust. The pump rod, therefore, works in tension, and can, if required, be conveniently extended to place the pumps below the prime mover.

The pump has been made in two sizes, 3-in. or 6-in. diameter cylinder, and can work either on town gas, producer gas, or, with a suitable carburetter, on petrol. It is stated that the 3-in. pump can lift 9200 gallons of water per hour to a height of 8 ft., or with different water ends, 575 gallons per hour to 172 ft. The 6-in. pump has an output varying from 46,800 gallons per hour at 12 ft. head to 1500 gallons at 500 ft.

In a modification of this type of pump which is being made in America the circular mass of solid metal is omitted and replaced by a vertical tube underneath, which has an inlet valve at the bottom of it. The action is similar to the pump just described. The upward working stroke produces an acceleration of the tube, plus the water contained in it, but the cushion formed by the compressed mixture above the piston arrests the upward movement of the piston and tube with a retardation which exceeds that of gravity. Consequently the water in the tube continues to flow upwards to the delivery, the tube only being arrested. The actual water delivered is nearly twice the theoretical displacement, and the smooth working is due to the peculiar acceleration and retardation which occur and which are possible only because there is no flywheel.

Oil Engine Types.—It is not nearly such a simple matter to classify oil engines as it is gas engines. The latter practically all work on the constant-volume cycle and are either four-stroke or two-stroke. Whatever gas is used, the mixture with air takes place before compression. There is no known case of pure gas being injected after compression. Compared to some types of oil engines, compression and maximum explosion pressures are low, and do not vary to any great

extent in the different types. Ignition in a gas engine is generally controlled from outside the cylinder, and is usually caused by an electric spark. Such considerations as these amount to saying that the great majority of gas engines are low compression, constant volume, electric ignition, with compression of the working substance. They differ, as has been seen, chiefly in mechanical design.

Oil, on the other hand, is a volatile substance which lends itself to a variety of methods of procuring and igniting the working substance in an oil engine.

If a "heavy" oil is to be used, which does not volatilise easily, it can be heated in a *vaporiser* and mixed with air before being introduced through an inlet valve into the cylinder. Any "light" oil or spirit, such as petrol, which evaporates at ordinary temperatures, may be mixed in a *carburetter* which does not require heating. This is the principle adopted in the petrol engine used for mechanical transport or for aviation.

Again, the vaporiser can be made to form an extension of, or even a part of, the combustion chamber of an oil engine. The oil can be injected into such a vaporiser either near the commencement of the suction stroke or near the end of compression. In the first case it then intermingles, in the form of oil vapour, with air which is drawn into the cylinder by the suction of the piston, and the working substance is compressed on the return stroke and ignited, usually by the heat of the vaporiser itself, when the engine has been properly started. The Hornsby-Akroyd is a good example of this type of engine. The compression pressure must be kept comparatively low, as in a gas engine, to prevent pre-ignition of the mixture before the end of the compression stroke. Such engines are often called low-compression oil engines. They are used chiefly for small power requirements (up to 100 B.H.P.), and the compression is between 50 and 100 lb. per sq. in., depending on the flash point of the oil used.

In the second case, when oil is injected near the end of the compression stroke, the danger of pre-ignition is avoided (unless the fuel valve leaks) and compression of the air alone may be carried considerably higher. The vaporiser is modified to form a "hot-bulb," and the oil is usually pulverised by specially designed nozzles as it enters. There are many examples of this type of engine which vary considerably in detail. They are often called hot-bulb engines, a term which is also occasionally applied to the low-compression type just mentioned. On account of the fact that only air is compressed they are also referred to as "semi-Diesels." The Petter oil engine was one of the first of this type.

Another type of oil engine employs no vaporiser at all, but relies on the atomising of the oil by other means. In this type the oil is sprayed into the cylinder at the end of compression, either by compressed air as in the Diesel engine, or by a pump and spring, as in the engines variously known as "cold-starting," "high-compression," or "solid-injection" oil engines. In these types no ignition device is required, the heat in the combustion chamber being sufficient to fire the mixture as soon as the oil is injected.

If oil engines were classified according to the way in which the

working substance is mechanically handled, they would fall very naturally into the three groups of low-compression, medium-compression, and high-compression, as typified by the Hornsby-Akroyd, the "cold-starting" crude oil engine, and the Diesel engine proper. The objection to such an arrangement lies in the fact that Diesel engines, which work on the constant-pressure cycle, are frequently considered in a class apart from other oil engines, most of which approximate to the constant volume cycle. The consequence of this is that the term high compression, which strictly belongs to the Diesel, has been applied in practice to "cold-starting" types, which are really medium compression, and which might more appropriately be named *sprayer* oil engines to destinguish them from those which require a hot-bulb as well. The complexity of types has led to a certain amount of looseness in the designation of the different kinds of oil engines, which makes it difficult to define them clearly. The classification adopted here is, therefore, one of convenience, but it should not cause confusion if the terms are limited to the sense attached to them.

(*a*) *Vaporiser Oil Engines* (low compression).—To cover all oil engines in which the oil is vaporised and drawn into the cylinder with air on the *suction stroke.* Which means that the working substance is compressed. Such an engine functions like a gas engine and has necessarily a low compression (50 to 100 lb. per sq. in.).

The vaporiser, however, may be external to or part of the compression space of the cylinder. In the former case it is replaced by a carburettor when no heating of the fuel is required.

(*b*) *Hot-bulb Oil Engines* (low compression) (semi-Diesel).—To refer to the type in which air only is compressed and the fuel injected near the end of the compression stroke and fired by the heat of a hot-bulb. The compression in this case may be higher than in (*a*) (often about 160 lb. per sq. in.), since pre-ignition troubles are absent. In most cases the engine works on the constant-volume cycle.

(*c*) *Sprayer Oil Engines* (medium compression) (cold-starting or solid injection).—In which air only is compressed, the oil being mechanically injected and atomised by a nozzle or sprayer near the end of the compression stroke. No hot bulb or vaporiser is used for ignition, the heat of the combustion chamber being just sufficient to fire the oil. The compression is necessarily higher (about 270–350 lb. per sq. in.), but burning takes place mostly at constant volume, the final pressure being about 550 lb. per sq. in.

(*d*) *Diesel Oil Engine* (high compression).—In which again air only is compressed, but in this case to about 500 lb. per sq. in., and the oil is injected through a pulverising nozzle by means of compressed air instead of mechanically. The mixture is fired by the heat of compression but burns at constant pressure.

As all these four types can be designed to work on the four-stroke or the two-stroke cycle, this distinction in the case of oil engines is better made as a sub-division.

(*a*) **The Dorman Petrol Engine.**—In order to keep this book within reasonable limits it has been decided not to make more than a passing reference to heat engines which are primarily designed for mechanical transport. One example of the petrol engine proper must

therefore suffice, and is shown in Fig. 100, by kind permission of the makers, Messrs. W. H. Dorman & Co., Ltd., of Stafford.

In this engine, air and petrol vapour are mixed in a carburettor (not shown) and pass through an induction pipe to each of four inlet valves in turn, shown on the right-hand top of the cylinder head in the end elevation. The engine works on the four-stroke cycle, and as there are four cylinders there are two impulses per revolution. Ignition is by sparking plugs placed just under the inlet valves and actuated by a high-tension magneto, which is gear-driven from the inlet valve camshaft. The products of combustion are removed through the exhaust valves shown on the left-hand top of the cylinder head, and these exhaust valves are operated from a second camshaft through tappet rods. Both camshafts are driven at half speed from the crankshaft by a single silent chain, the gears for which can be seen on the left of the side elevation.

The cylinders are water-jacketed on the sides and on the top, the circulation being maintained in this case by "thermo-syphon" assisted by a small water "impeller," seen in section to the left of the cylinder barrels and driven by a leather "V" belt from a pulley carried on an extension of the exhaust-valve camshaft. Water from the bottom of the radiator (not shown) enters the cooling system through this "impeller," passes round the cylinder liners, and then back across the cylinder heads and past the valve boxes to the outlet pipe seen at the top left-hand corner. A rubber connection is made between this water-outlet pipe and the top of the radiator, in which the water gradually gives up its heat as it passes down to the bottom. An air fan can be fitted on the outside of the "impeller" spindle to assist in this cooling effect.

The engine is started by means of the crank handle on the left, which can be clutched to the crankshaft by pushing it inwards against a spring. As soon as the engine gets away and the crankshaft is travelling faster than the starting handle this clutch automatically disengages itself.

The particular engine illustrated develops 13½ B.H.P. when running at 1000 R.P.M. Each cylinder is 69 mm. bore by 120 mm. stroke. Mechanical features of this design may be noted. Each of the four cranks is balanced, the two middle ones by a single balance weight shown in between. This makes for smooth running and prevents transverse oscillations of the crankshaft. The cylinders consist of cast-iron liners pressed into a one-piece aluminium casting, which forms the jackets and the crank case. The lower joint is made watertight by a rubber ring held in compression, which also allows for any difference in expansion between the two metals. Lubrication of the interior is maintained by a small oil pump (not shown) driven from a camshaft by gearing which can be seen just to the right of the chain drive. This pump circulates oil to the flywheel main bearing and to the dipper troughs under each connecting rod, which are made higher on one side than the other, so that they shall carry a reasonable amount of oil when the engine is going uphill. An extension on each connecting rod dips into these troughs and splashes the oil all over the crank case. A pipe connection, which can be seen on the right of the side elevation, behind

the exhaust outlet, allows oil vapour to rise from the crank chamber to

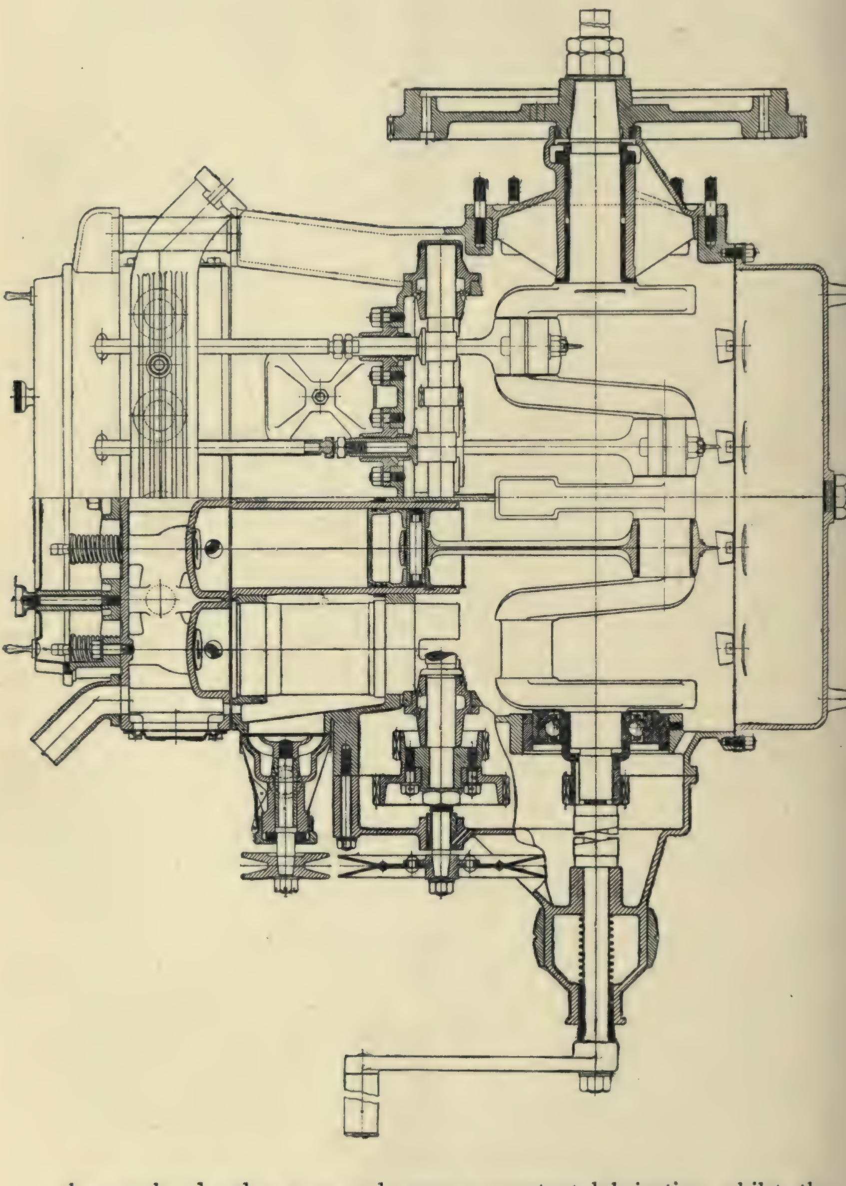


FIG. 100.—Four-cylinder petrol engine (Dorman).

the overhead valve gear and ensures constant lubrication whilst the engine is running.

By far the largest number of engines of this class are designed to

run on petrol,[1] which limits the compression pressure to about 80 lb. per sq. in. It is, however, quite possible to fit a vaporiser instead of a carburettor and use such oils as paraffin (kerosene).[2]

The Priestman Oil Engine.—This engine, although no longer made with a separate vaporiser, forms an important link in the development of the oil engine, and is typical of the case where a four-stroke

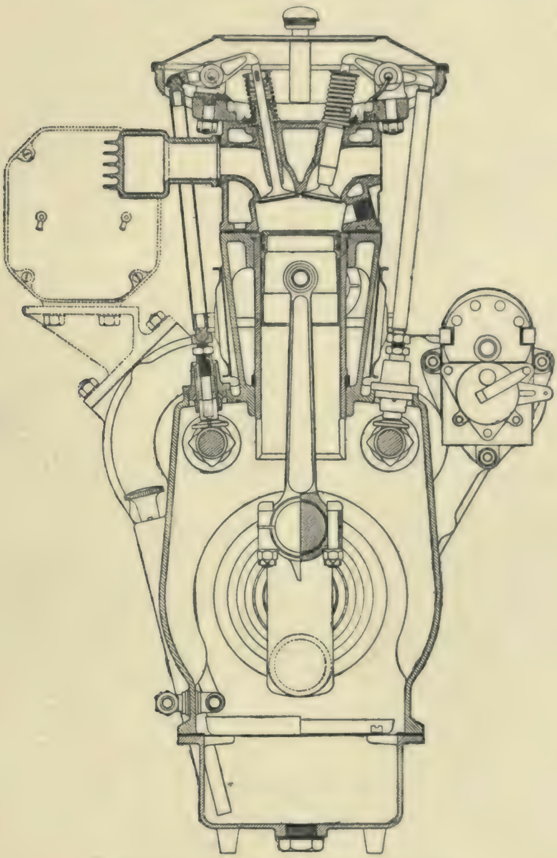


FIG. 100A.—End elevation of Fig. 100.

cycle was obtained with heavy oils, vaporised separately, mixed with air and introduced through an inlet valve to the cylinder. It was, in fact, the first successful heavy oil engine, and was produced in 1888. The vaporiser was placed underneath the cylinder in the engine bed, a jet of oil was mixed with air and sprayed into this vaporiser, where the

[1] For detailed information of fuels suitable for carburettors, see "Liquid Fuels for Internal-Combustion Engines," by H. Moore, 1918, chap. viii.

[2] *Ibid.* chap. ix.

mixture was heated by a jacket through which the exhaust gases were passed. On the suction stroke of the piston the inlet valve opened and drew in this mixture with additional air into the cylinder, to be compressed and fired by electric ignition in the ordinary way. The design was clearly an adaption of the principle of the gas engine to the use of heavy oil mixtures. An early type of this engine is fully described with drawings and tests by Professor W. C. Unwin, in the *Proceedings of the Institution of Civil Engineers*, vol. cix. (1892), p. 1.

The Hornsby-Akroyd Oil Engine.—This engine was the first of the type in which a vaporiser (sometimes called a hot-bulb) formed part of the compression space of the cylinder. It was invented by Mr. Akroyd Stuart before 1890. A sectional diagram of one of the

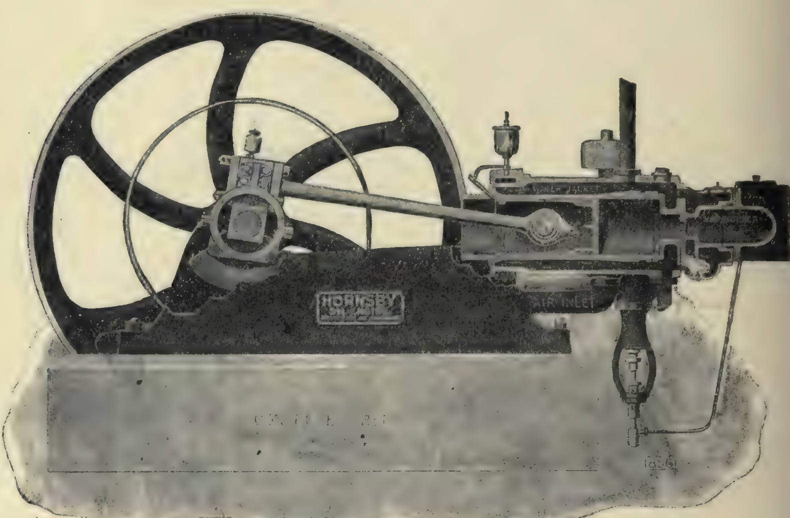


FIG. 101.—Vaporiser oil engine (Hornsby-Akroyd).

earliest designs is shown in Fig. 101, as made by Messrs. R. Hornsby & Sons, Ltd., of Grantham (now Messrs. Ruston & Hornsby, Ltd.).

At the beginning of the suction stroke the oil is injected by means of the small pump into one side of the vaporiser, which can be seen connected to the end of the cylinder. At the same time air is drawn in direct to the cylinder through an air valve and a port, which is also used for the products of combustion to pass to the exhaust valve in the same valve chest. This valve chest is on the far side of the engine as shown, the two valves being operated by rocking levers in the larger sizes, which are cam-driven from an intermediate lay shaft. The engine works on the four-stroke cycle and is water jacketed as much as possible. The end of the vaporiser is left unjacketed, however, to enable it to be heated up by a separate lamp for starting purposes. When once running ignition is automatic. One or two flywheels are

fitted, according to the size and duty, and a large number of this type of engine have been made in the past up to about 66 B.H.P.

(*b*) **The Petter Crude Oil Engine.**—This engine may be taken as typical of a considerable number of hot-bulb crude oil engines, which are often referred to as "semi-Diesels." They work, however, on the constant-volume cycle, and differ from what are here called vaporiser engines, in that air only is compressed before the fuel is injected. Fig. 102, reproduced by kind permission of Messrs. Vickers-Petters, Ltd., of Ipswich, shows one cylinder of a vertical hot-bulb oil engine. It works on the two-stroke cycle, so that there is one impulse per cylinder per revolution.

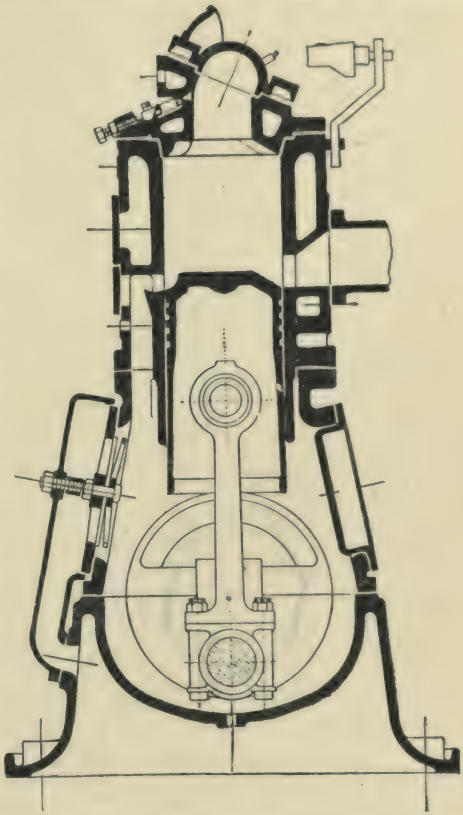


FIG. 102.—Hot-bulb oil engine (Petter).

Automatic air-inlet valves are placed on the side of the crank case, and air is drawn in by the upward stroke of the piston. As the piston descends this air in the crank case is compressed, about 5 lb. above atmosphere, to force it into the cylinder through the air ports, seen on the left, as these ports are uncovered by the piston at the end of the downward stroke. The air in the cylinder is then compressed to between 160 and 180 lb. per sq. in., and the fuel injected near the end of the compression stroke into the hot bulb shown on the top of the cylinder. The explosion pressure reaches about 290 lb. per sq. in., and the products of combustion are swept out as the exhaust ports, seen on the right, are uncovered by the downward motion of the piston. In the other types of hot-bulb engines a water drip was often used to prevent pre-ignition, but designers have now had sufficient experience to eliminate this feature.

A four-cylinder vertical hot-bulb oil engine of this type is described, with illustrations, in the *Engineer*.[1] This engine has four cranks, Nos. 1 and 2 opposite to one another, and Nos. 3 and 4 opposite one another and

[1] *Engineer*, vol. cxxi. (1916), p. 234.

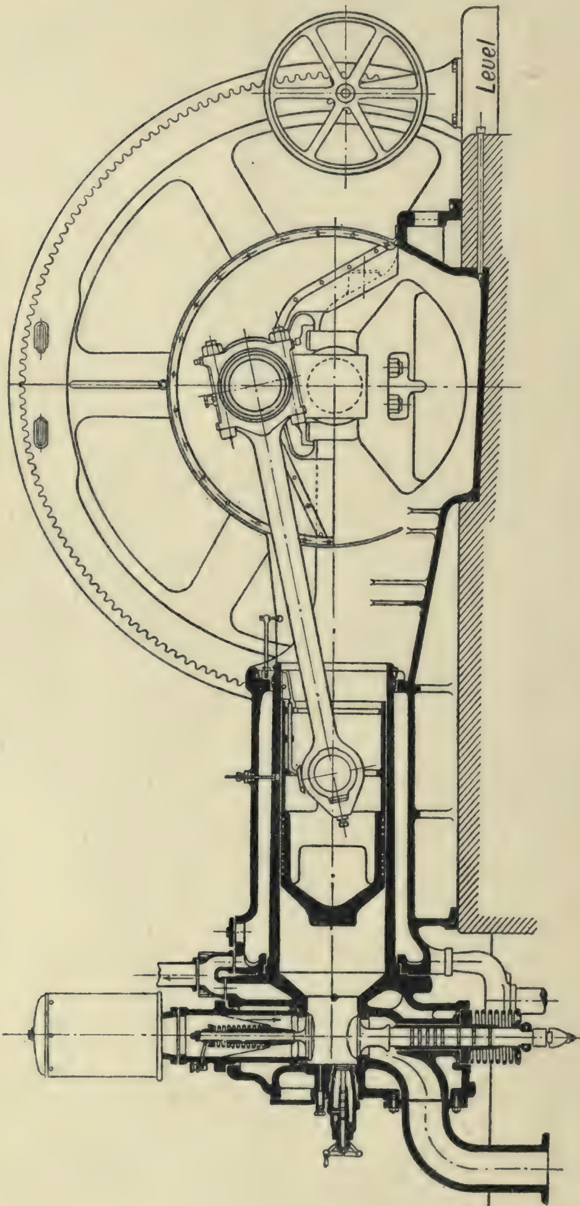


FIG. 103.—Sprayer oil engine (Crossley).

at right angles to the first pair. The cylinders are 16 in. diameter by 18 in. stroke, and the engine develops 300 B.H.P. when running at 250 R.P.M. The fuel consumption need not exceed 0·45 lb. of oil (sp. gr. 0·92) per B.H.P. per hour. A separate pump is used for each cylinder, but in this engine all four pumps are controlled by a single governor, which alters both the time of opening and the stroke of the pumps by an ingenious mechanical arrangement. The lubrication of the gudgeon pin may be noted. It is hollow, with a spring-loaded scraper at one end, which picks up the oil and guides it into holes which spread it over the pin surface.

(*c*) **Sprayer Oil Engines** (medium compression).—In this type of oil engine, which has been developed simultaneously by a number of makers, no hot bulb or vaporiser is used, but the oil is atomised by a mechanically operated sprayer to such an extent that the fine oil-mist intermingles with the air at the end of the compression stroke sufficiently to cause explosion mainly at constant volume. No ignition device is necessary, the air being compressed up to about 300 lb. per sq. in., which is found just sufficient to fire the mixture.

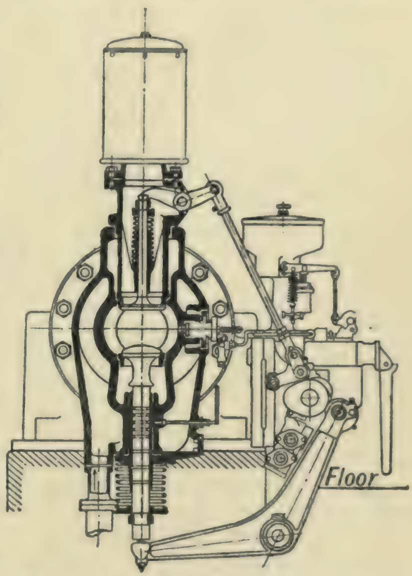


FIG. 103A.—End elevation of Fig. 103.

The *Crossley* oil engine of this type is shown in Fig. 103, and is reproduced here by the courtesy of the firm and of the editor of the *Engineer*.[1] The engine works on the four-stroke cycle, the air-inlet valve being above and the mechanical oil sprayer being on the right-hand side. At the end of the compression stroke oil is forced through specially designed nozzles by a spring-loaded spindle, which is lifted by a pump driven off the cam shaft. The spring is adjusted to give a fuel injection pressure of about 1200 lb. per sq. in.

The piston is so designed that the last inch of the compression stroke forces a comparatively large volume of air through a narrow annular space formed by the plunger end on the piston and the entrance to the combustion chamber. This sets up considerable turbulence in the combustion chamber, which helps to ensure a thorough mixture of air and fuel. The engine illustrated has a cylinder $18\frac{1}{2}$ in. in diameter by 28 in. stroke, and a normal rating of 120 B.H.P. when running at 180 R.P.M. Quantity governing is used, the surplus oil being bye-passed back to the pump suction through a valve controlled by a centrifugal

[1] *Engineer*, vol. cxxviii. (1919), p. 252.

type of governor. Almost any kind of crude oil can be utilised, from paraffin to tar oil.

This figure should be compared with the design of a modern gas engine as shown in Fig. 93. The similarity will at once be noticed, showing how the original idea as typified in the Priestman oil engine of making oil engines like gas engines is being reverted to. This bears out the contention that the gas engine and the oil engine, at any rate as far as the horizontal four-stroke type is concerned, need not differ in mechanical design except in so far as the arrangements for injecting and mixing the working substances are concerned.

Tests by Professor Burstall on this engine show that with residual petroleum having a net calorific value of 18,000 B.Th.U. a consumption of 0·425 lb. of oil per B.H.P. per hour was obtained when the engine was developing 125 H.P. This gave a brake thermal efficiency of 33·26 on the "air-standard." The engine is started by compressed air, which is admitted through the valve shown on the left of the side elevation. No preliminary heating is required, which has led to this type being called the "cold-starting" engine.

An engine of a similar type is made by Messrs. Ruston & Hornsby,[1] Ltd., at Lincoln, and an independent test by Capt. Riall Sankey [2] shows that with oil of a lower calorific value of 18,050 B.Th.U. per lb. a consumption of only 0·401 lb. of oil per B.H.P. per hour was obtained at full load on engine with two side-by-side cylinders each 18½ in. diameter by 30 in. stroke, and developing 236 B.H.P. when running at 175 R.P.M. This performance represents a brake thermal efficiency of 35 per cent. compared to the "air-standard."

The Doxford Oil Engine.—Although some classification might place it as a modified hot-bulb engine, the *Doxford* opposed piston "solid-injection" oil engine must be considered as an interesting development of the sprayer oil engine. In this engine, which works on the two-stroke cycle, there are two opposed pistons moving outwards from the centre, on the principle of the Oechelhauser gas engine and the Junkers-Diesel. It differs from the latter, however, in not using compressed air to inject the heavy oil, but a mechanically operated and controlled sprayer valve, as seen in Fig. 104, supplied by the courtesy of William Doxford & Sons, Ltd., of Sunderland. Two of these valves are used on opposite sides, but placed one above the other so as to cover as large amount of the combustion space as possible. The fuel oil, compressed by an ingenious arrangement to about 8000 lb. per. sq. in., is led along the small bore pipe to both ends of the valve spindle. The light spring at the far end of the spindle is only used to keep the valve in position when the engine is at rest. The valve is opened and shut by a bell-crank lever, whose pivot position may be altered to reverse and control the engine, which in this case is designed for marine propulsion. The compression pressure is about 300 lb. per sq. in., and the maximum explosion pressure about 500 lb. per sq. in. Ignition is assisted by specially designed pistons, which carry false crowns made of steel forgings, as can

[1] For description with details of oil sprayer, see *Engineering*, vol. cx. (1920), p. 8.
[2] *Engineer*, vol. cxxx. (1920), p. 506.

be seen from Fig. 105.[1] The inside of these pistons as well as the under side of the crown are water-cooled, but a space is left between the piston and its head, except where the two are bolted together, and the temperature of the outer face of the crown reaches and remains at about 1000° F.

A test of a Doxford sprayer oil engine of this type designed for 500 S.H.P. at 112 R.P.M. is reported in the *Engineer*,[2] and shows a consumption of 0·422 lb. of oil per S.H.P. per hour when running at normal load on Mexican fuel oil of 0·9 sp. gr. The injection oil pressure

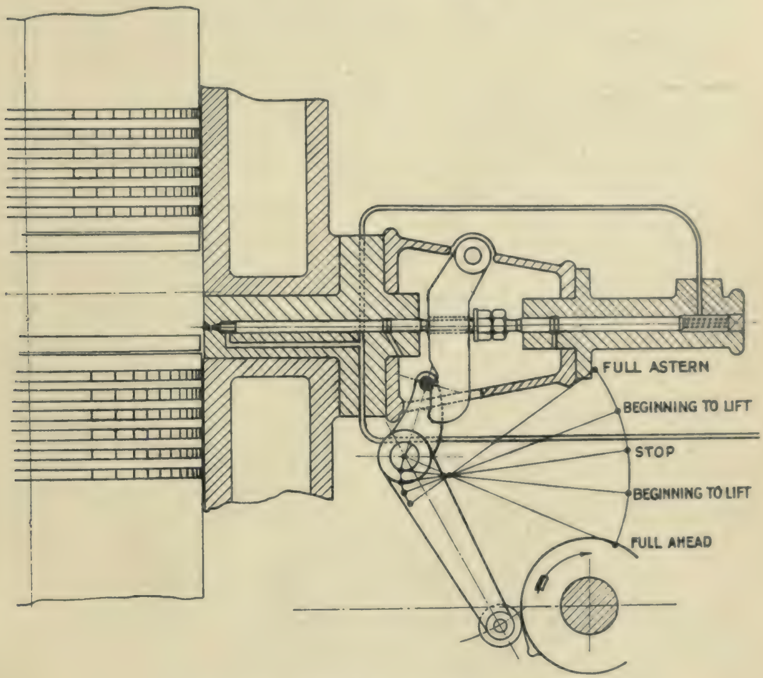


FIG. 104.—Sprayer for Doxford oil engine.

was 8200 lb. per sq. in. and the mean effective pressure 92 lb. per sq. in.

The Still Engine, which combines internal combustion on one side of a piston with steam on the other, has been carried out experimentally in several forms,[3] but lends itself very naturally to the use of the sprayer type of oil engine.

A Still engine of this type is reproduced in Fig. 106 by the courtesy

[1] Reproduced from British Patent Specification No. 124,534, accepted March 18, 1912.

[2] *Engineer*, vol. cxxviii. (1919), p. 64.

[3] See paper by F. D. Acland, Royal Society of Arts, May 26, 1919, reprinted in the *Engineer*, vol. cxxvii. (1919), p. 540.

of the Council of the Institution of Naval Architects.[1] Like the Doxford engine just described, it is built with opposed pistons on the Junkers principle and the internal combustion end works on the two-stroke cycle. The engine shown, however, is not a Diesel but works on the solid-injection principle. The oil fuel is sprayed in between the two pistons when they are together, and the exhaust leaves through ports uncovered by the lower piston at the end of its stroke. From here the exhaust gases enter a generator, which is something more than a feed-water heater, and pass round tubes containing water from the main feed pump. The boiling water leaving this generator is passed round the jacket of the central or internal-combustion part of the cylinder, where it receives more heat from the internal-combustion

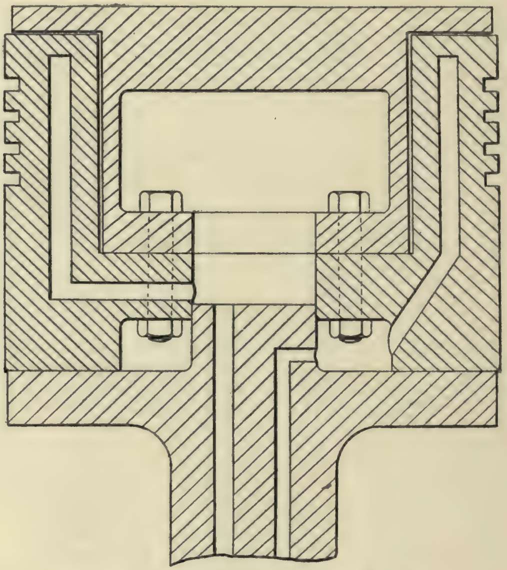


FIG. 105.—Piston for Doxford oil engine.

portion of the engine, and in doing so helps to keep the cylinder walls cool. From there it is conducted to a boiler, shown diagrammatically, and converted entirely into steam.

The outer ends of each piston form a steam cylinder, into which this steam is passed to act on the return stroke of the engine. When the engine is running under normal conditions sufficient steam is generated from the heat of the exhaust gases and from the heat transferred in the cylinder jacket to run the steam ends of the engine without any firing of the boiler. By fitting oil burners, however, extra steam is available for overloads, whilst it also forms a convenient method of starting up the engine from cold.

[1] Paper by W. Denny, "Still" and "Sulzer" Marine Engines, read before *Inst. N.A.* July 8, 1920. Reprinted in *Engineering*, vol. cx. (1920), p. 98.

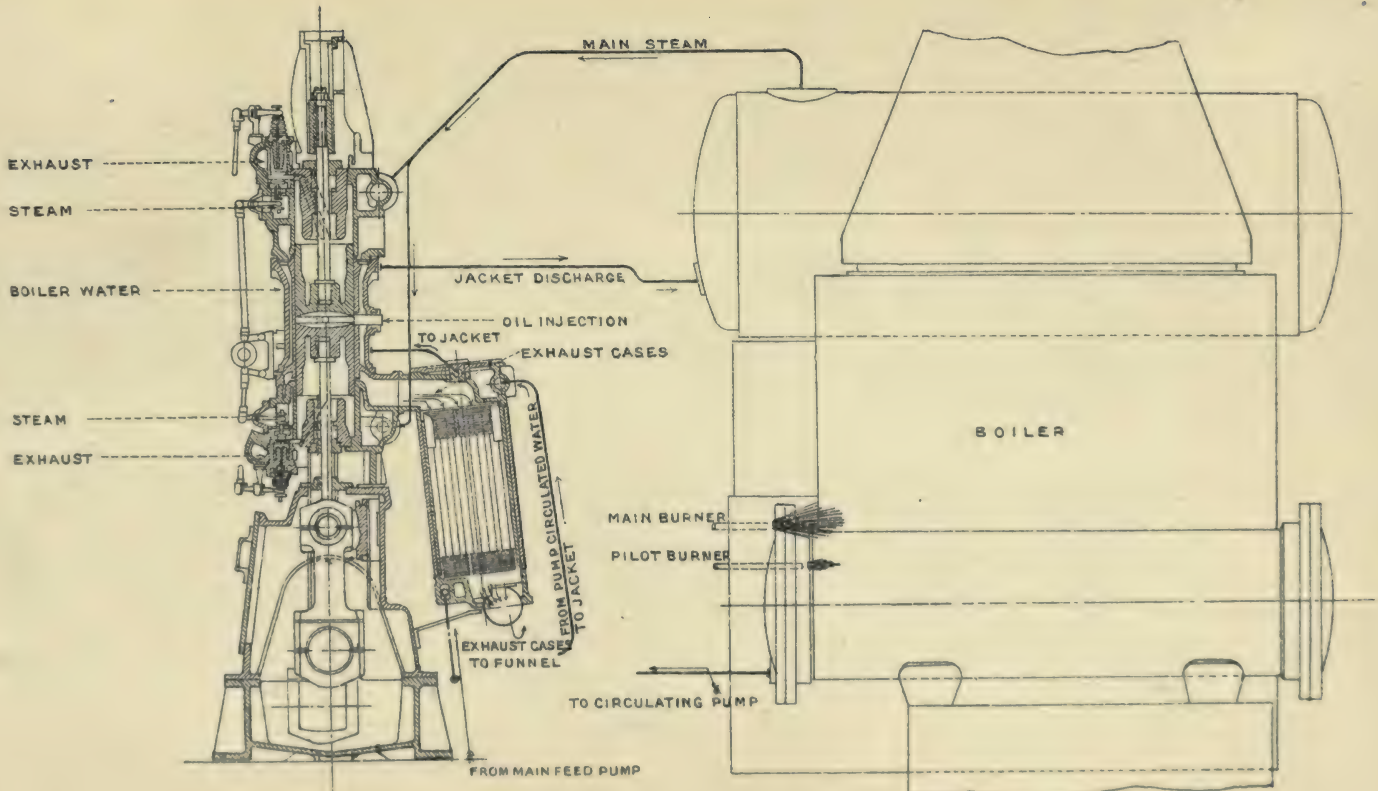


Fig. 106.—" Still " engine.

The engine illustrated was an experimental one with four cylinders each $7\frac{1}{2}$ in. diameter by 15 in. stroke, and developed 125 S.H.P. when running at 250 R.P.M.

The accompanying indicator cards, Fig. 107, reproduced from the same paper, show that the compression pressure of the internal combustion end was about 435 lb. per sq. in. and the maximum pressure about 575 lb. per sq. in. above atmosphere. As can be seen the steam ends were in this case condensing.

In the trials described two of these engines were run in an experimental twin-screw ship. The fuel used was shale oil (0·85 sp. gr.), and showed a total consumption of only 0·42 lb. per S.H.P. at normal full load.

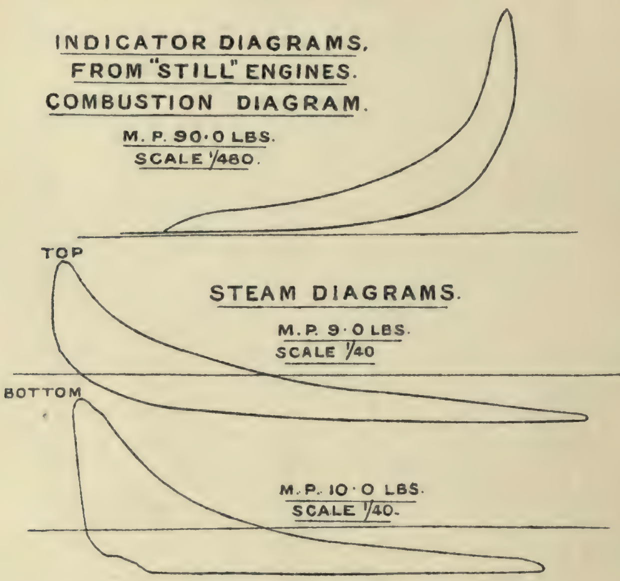


FIG. 107.—Indicator cards from a "Still" engine.

(*d*) **The Diesel Oil Engine** (high compression).—This type of engine, designed from theoretical considerations by Dr. R. Diesel, was first made successfully in the year 1897, by the M.A.N. on the continent and by the firm of Mirrlees, in Glasgow. The progress between that year and 1912 is very clearly set out in a paper by Dr. Diesel read before the Institution of Mechanical Engineers.[1] Although large numbers of this type have been made and a considerable literature devoted to the subject, it may be stated broadly that developments since that date have been more in detail than in any fundamental departure from Diesel's design.

The engine is practically always constructed with single-acting vertical cylinders placed side by side, but can be designed to work on either the four-stroke or the two-stroke cycle.

[1] *Proc. Inst. Mech. Eng.* (1912), Parts 1 and 2, pp. 179 *et seq.*

An example of the *four-stroke vertical type* with trunk pistons is shown in Fig. 108, which is reproduced by the courtesy of the makers,

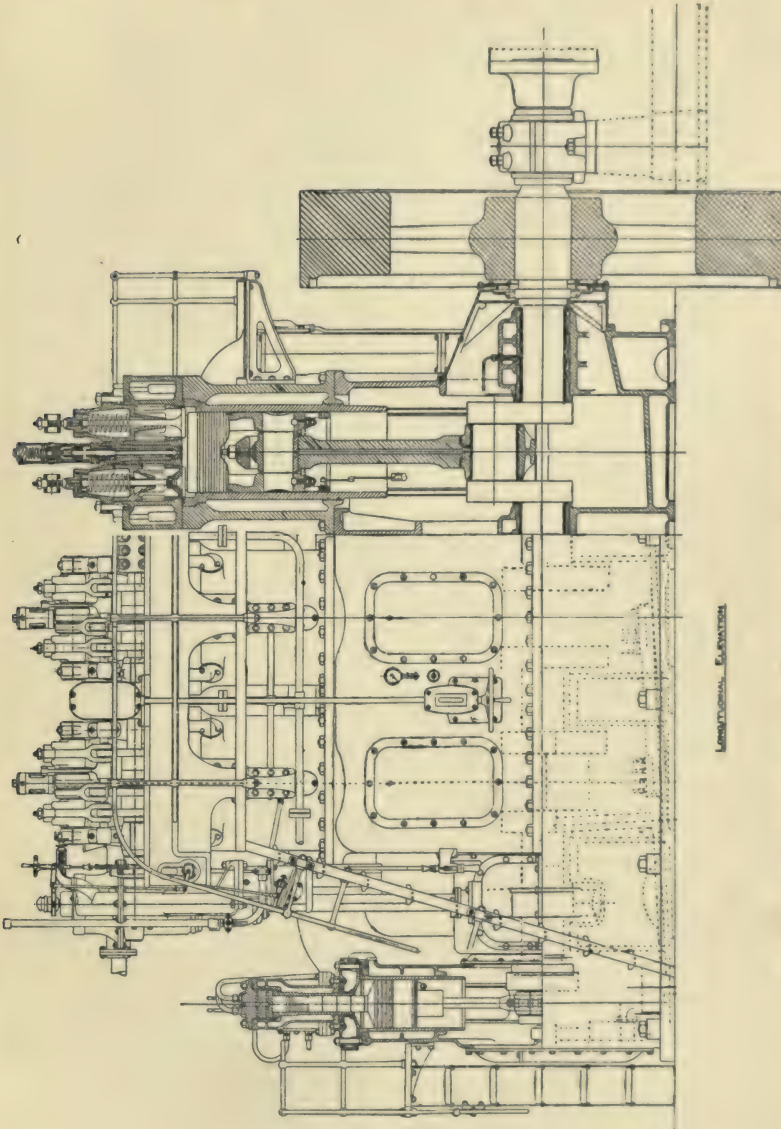


FIG. 108.—Four-stroke Diesel oil engine with trunk pistons (Mirrlees, Bickerton & Day).

Messrs. Mirrlees, Bickerton & Day, ot Stockport, near Manchester. In this size each cylinder is 20 in. diameter by 24 in. stroke, and each

develops 125 B.H.P. when running at speeds of between 200 and 250 R.P.M. Three valves are fitted in the top of the cylinder. The right-hand one is for air, in the centre is the fuel valve, and on the left the exhaust valve. All three are operated by rocker levers, whose movements are controlled by revolving cams on a layshaft driven through gearing from the crankshaft.[1]

Air only is drawn in on the suction stroke and compressed to about 500 lb. per sq. in. or a temperature round about 800° C. (abs.). An air compressor, driven off the left-hand end of the crankshaft, is used to supply air under pressure of between 700 and 800 lb. per sq. in. to the fuel valve, which opens at the end of the compression stroke. This air injects and intermingles with the oil, which is ignited solely by the temperature of compression. The amount of fuel used per stroke is controlled by small fuel pumps, which are regulated by a governor. Combustion takes place more or less at constant pressure, giving the characteristic flat top to the Diesel indicator card, and the burnt gases are swept out through the exhaust valve on the first return stroke. The engine is quite as compact as a high-speed steam engine, though built on heavier lines. It is totally inclosed, with forced lubrication throughout. The mean effective pressure is 85–90 lb. per sq. in. on I.H.P. basis. The same firm also built four-stroke vertical Diesels with crossheads instead of trunk pistons.[2] This is a reversion to the original engine of 1897, and is becoming more general again, particularly for marine work.

Although more head room is required, and the engine is therefore more expensive, the mechanical advantages of the crosshead type are considerable. Side thrust between the piston and the cylinder walls is eliminated and the wear and tear considerably reduced. Smaller piston clearances can be used without the same danger of seizing, which keep the piston more air-tight under pressure and prevent possible scouring of the cylinder walls. The piston itself no longer acts as a guide, and there is no gudgeon pin inside it, so that lubrication is simpler and more reliable. The design approximates even more closely to the vertical steam engine with which most makers are familiar, so that experience gained here can be utilised to a greater extent.

Although not so common, except for very large sizes, the two-stroke cycle is equally applicable to the principle upon which the Diesel engine works.

A sectional end elevation of the two-stroke type is shown in Fig. 109, which is reproduced from *Engineering*[3] by the kind permission of the editor and of the maker, F. Tosi, of Legnano, in Italy.

Although it embodies the crosshead type, it will be noticed that advantage has been taken of the extra height to save length by placing the air compressor and scavenging pumps behind the cylinders and driving them by means of rocking links from the connecting rod. The engine

[1] For various methods of operating Diesel engine fuel valves, see two articles by Koenemann in the *Zeit. Ver. Deut. Ing.* vol. lx. (1916), pp. 997 and 1077.

[2] A description of a 350 H.P. crosshead type Diesel may be seen in the *Engineer*, vol. cxxii. (1916), p. 414.

[3] *Engineering*, vol. xciii. (1912), p. 696.

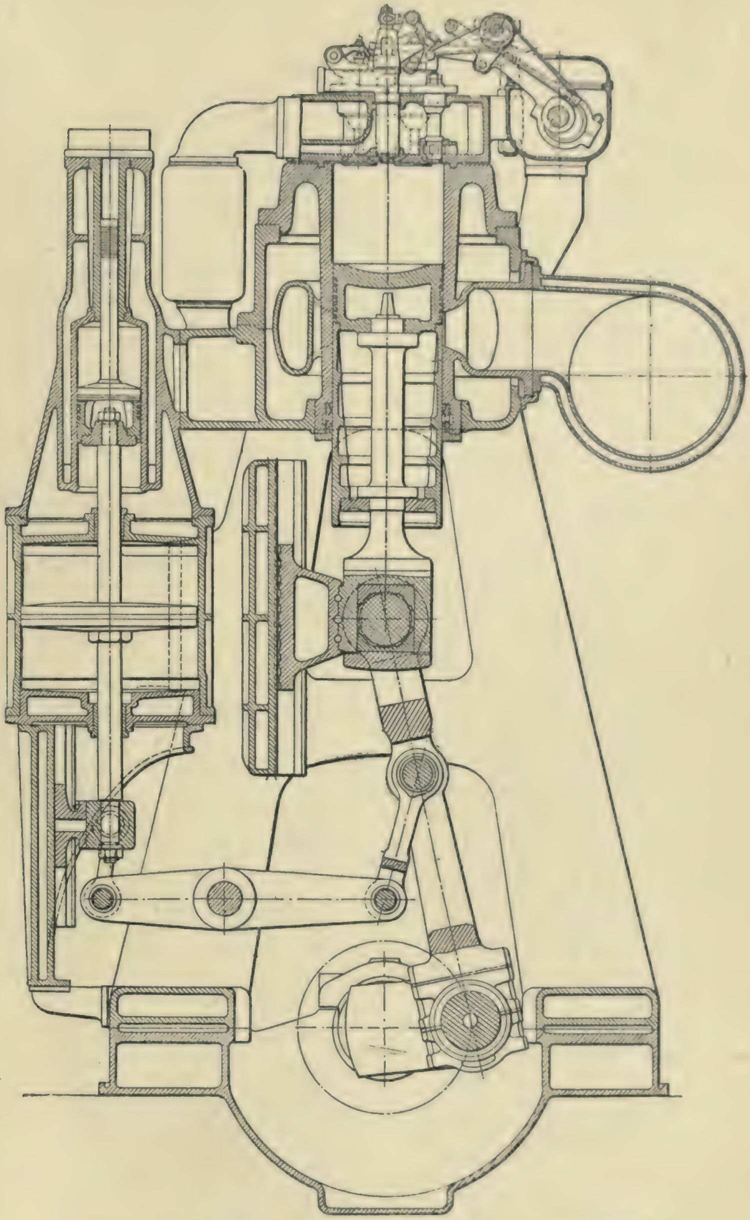


FIG. 109.—Two-stroke Diesel oil engine, crosshead type (Tosi).

illustrated develops 500 B.H.P. at 170 R.P.M. in four single-acting cylinders with water-cooled pistons. The scavenging pumps, necessary for two-stroke operation, are in this case two in number. They are double-acting, and one of them can be seen just to the left of the main crosshead. The three-stroke air compressor is mounted on an extension of the scavenging pump rods. There are two low-pressure cylinders, and the intermediate and high-pressure stages are carried one above each of these cylinders. The compressor is well water-jacketed, and intercoolers (not shown) are provided to keep down the temperature of the compressed air below that which might fire the lubricating oil.

Just after the piston has uncovered the exhaust posts at the bottom of a working stroke the scavenging air is driven through the pipe, seen between the air compressor and engine, to two valves dotted at the top of the cylinder on each side of the fuel valve. When these valves open the air rushes in and sweeps out the burnt gases into the annular exhaust and exhaust pipe seen on the right. On the return stroke the piston soon overruns these exhaust passages and compresses the remaining air to about 500 lb. per sq. in. As in the four-stroke type, the fuel is fed to the fuel valve through small oil pumps, of which there is one for each cylinder, and the quantity passing through the pumps is controlled by a centrifugal governor. The scavenging air-valve rocking-levers are cam-driven, but in this particular make the fuel valve is operated by a rod driven by an eccentric. The engine is designed for marine work, and by mounting this rod on a swinging pivot the speed can be varied by altering the lift of the valve and reducing the time of injection and the angle of advance. It also enables the engine to be reversed for running the ship astern. The small valve on the right-hand top of the piston is for the starting air. This firm, whilst still retaining the crosshead type, are reverting to the four-stroke cycle, particularly in large sizes.[1]

The Junkers Two-stroke Diesel, with opposed pistons.—This type of engine, which is an adaptation of the principle of the Oechelhauser gas engine to Diesels, is referred to in Dr. Diesel's paper of 1912,[2] and so far has been chiefly developed by Continental firms, such as Gebr. Klein, A.E.G., and Nobel. In this country the Doxford opposed piston sprayer oil engine (see p. 228) embodies similar mechanical features, and the Camellaird-Fullager Diesel (p. 240) may be said to be a development of the type.

An example of a Junkers Diesel can be seen in Fig. 110. It is reproduced from the *Zeitschrift des Vereines deutscher Ingeneure* by the kind permission of the editor.[3] This engine, which was built to the inventor's designs, has three cylinders each 200 mm. ($7\frac{1}{2}$ in.) in diameter with 700 mm. (27 in.) stroke, and develops 67 B.H.P. per cylinder when running at 250 R.P.M. A double-acting scavenging pump with automatic valves and a three-stage air compressor are driven off an extension of the crankshaft. The fuel injection valve is driven by an eccentric and has two specially designed nozzles, which spray the oil

[1] For a description of a six-cylinder four-cycle Tosi-Diesel, see *Engineering*, vol. cvii. (1919), p. 25.

[2] *Proc. Inst. Mech. Eng.* (1912), Parts 1 and 2, p, 199.

[3] *Zeit. d. Verein. deutsch. Ing.* vol. lxi. (1917), p. 283.

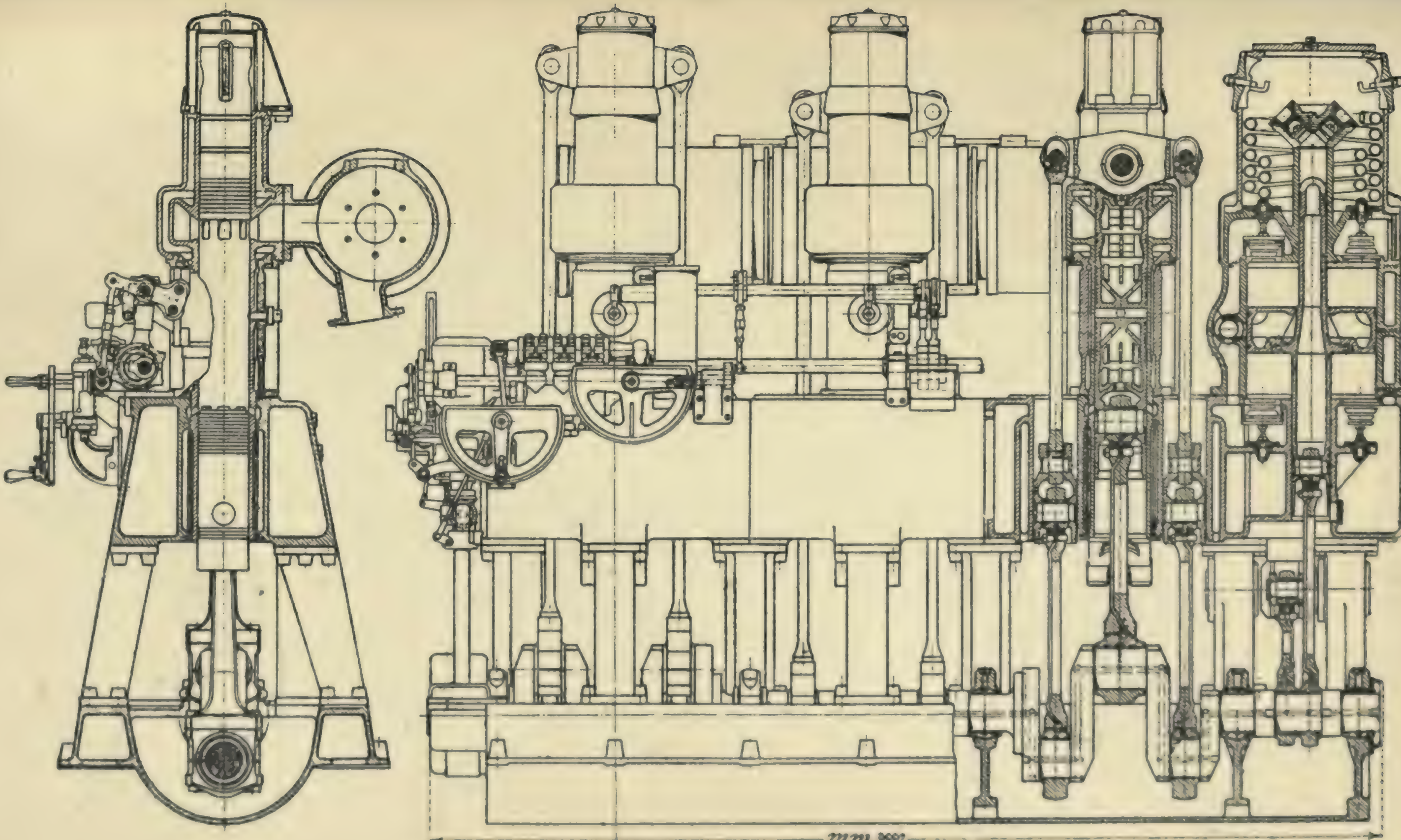


FIG. 110.—Junkers Diesel oil engine.

and air fanwise across the central combustion space when the two pistons are close together. As the two pistons draw apart they suck the intermediate air through these fan-shaped oil sprays and ensure a better combustion. The scavenging is positive and in one direction only. Towards the end of the outward stroke the upper piston uncovers exhaust ports first, and at the very end of the stroke the smaller air-scavenging ports are opened by the lower piston. The air then rushes in at a pressure of about 3 lb. per sq. in. above atmosphere and sweeps the remainder of the products of combustion through the exhaust. As the air ports close before the exhaust there can be no supercharging. The compression ratio is very high, 1 to 16, which gives a maximum pressure at the end of compression of about 700 lb. per sq. in. The average mean effective pressure of this type of engine may be taken at 155 lb. per sq. in. A back-pressure valve is fitted opposite the fuel valves. The starting in this case is effected by compressed air at about 1250 lb. per sq. in., which is passed through the fuel-injection valve, and the engine can start from cold on coal-tar oil. In the larger Junkers a separate starting valve is fitted as well as two fuel valves per cylinder. The whole control is centred in a single hand lever, which enables the engine to be manœuvred, ahead or astern, fast or slow, almost as well as a steam engine. The gear operated by this lever changes the period of injection, the angle of advance, and the lift of the fuel-valve needle. Each load requires a corresponding weight of air in the cylinder. This weight of air is directly proportional to its absolute pressure, and in some Junkers engines arrangements are made for partially throttling the exhaust, with the result that the scavenging air is left in the cylinder at a higher pressure, and more fuel can be burnt per stroke. Additional storage space for the extra amount of air is necessary, but temporary overloads of 50 per cent. have been met in this way. It should be remembered, however, that every slight increase in the pressure of the air before compression enormously increases the maximum pressure at the end of compression, and some form of safety release valve would be necessary to safeguard the cylinder with a 50 per cent. overload; for instance, the mean effective pressure on an I.H.P. basis is raised from 155 to as much as 230 lb. per sq. in., and the compression pressure to 1060 lb. per sq. in.

The mechanical construction of this type of engine has the one disadvantage of three connecting rods and crank pins for each cylinder, but against this must be set the fact that the system of opposed pistons practically balances the inertia forces of the reciprocating parts and that no cylinder covers are required. The cylinder itself is simply a long C.I. liner, reinforced by a steel casting round the combustion chamber in the centre with the water jacket in between. The pistons themselves are fitted with Junkers system of "pendulum" cooling. The explosion end of each piston is enclosed and half filled with liquid. The reciprocating movement of the piston flings the liquid against the inside of the piston head and the heat thus extracted is carried by the liquid on the return stroke to cooling fins, whence it passes to the cylinder walls and the water jacket. Junkers has also built this type of engine with horizontal cylinders up to 1000 B.H.P. at 180 R.P.M.

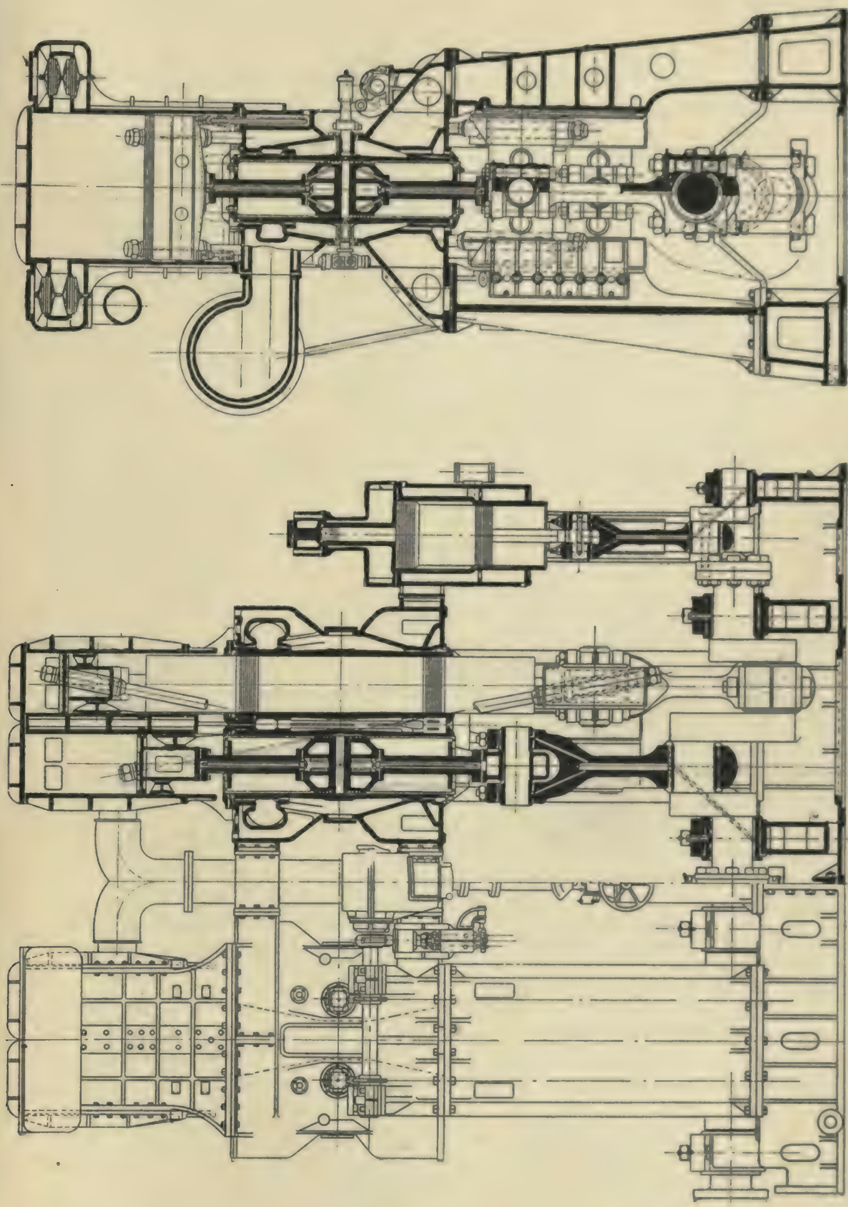


FIG. 111.—Cammellaid-Fullager Diesel.

in two cylinders,[1] which are reported to have given comsumptions of 176 gm. per B.H.P. per hour with oil of a lower calorific value of 10,000 cal. per kg. This corresponds to 0·387 lb. of oil per B.H.P. per hour with a calorific value of 18,000 B.Th.U. per lb.

Cammellaird-Fullagar Diesel.—In an instructive paper on the "Balance of Internal-combustion Engines,"[2] H. F. Fullagar shows that in order to obtain an engine with perfect balance, that is, so to dispose of all the forces and stresses set up within an engine that the crankshaft receives nothing but direct, positive, and useful impulses, it is necessary not only to achieve both primary and secondary balance of the moving parts,[3] but also to obtain balance of impulse. In single-acting engines the pressures on the pistons during the working and compression strokes set up reversals of stress in the crankshaft, whilst the reaction of the crosshead guides, or the side thrust of a trunk piston within the cylinder, as the case may be, also changes sign with each stroke.

The Oechelhauser gas engine or the Junkers Diesel is balanced for primary and secondary forces but not for impulse, unless four pairs of pistons are arranged in tandem on two sets of three cranks each at 90°. This necessitates a very long or very high engine, and Fullagar overcomes this drawback by arranging two pairs of pistons in twin and rigidly connecting the top piston of one cylinder to the lower piston in the other, and *vice versâ*. By placing two such sets on one crankshaft with the crank pairs at 90° perfect balance in all respects is obtained. The engine has the further advantage that only one connecting rod and crank pin is required for each pair of pistons instead of three as in the Junkers.

The explosions in four cylinders acting on four pairs of pistons produce eight impulses per revolution, which are transmitted to the crankshaft in equal pairs. An engine of this type, as made by the shipbuilding firm of Cammell Laird & Co., Ltd., of Birkenhead, is shown in Fig. 111, by the courtesy of the firm and of the editor of the *Engineer*.[4]

This represents a marine engine designed to develop 1000 B.H.P. at 110 R.P.M. in two pairs of cylinders. Each cylinder is 18½ in. diameter by 2 × 25 in. stroke, and the general action of the cycle is the same as in the Junkers Diesel. The pistons in this case are oil cooled and forced lubrication is used throughout. The fuel consumption of such an engine should prove very good, in view of some tests carried out by the firm on a totally enclosed experimental engine with two cylinders 11½ in. bore by 2 × 12 in. stroke. This engine under a continuous 72 hours' full-load trial averaged 0·42 lb. of oil per B.H.P. per hour when developing 295 B.H.P. at 350 R.P.M.

The positive scavenge enables very low air pressures to be used, the scavenging pressure being only 1½ lb. per sq. in. above atmosphere. This is an advantage inherent in the type of two-stroke cycle engines with opposed pistons, and reduces the power required for the scavenging pumps to a minimum.

[1] "The Junkers Engine," by Philip L. Scott, American Society of Automotive Engineers, reprinted in the *Automobile Engineer*, vol. viii. (1918), p. 7.

[2] *Proc. Inst. Mech. Eng.* (1914), Parts 3 and 4, p. 559.

[3] For primary and secondary balancing, see Inchley, "Theory of Heat Engines" (1920), ch. xx. p. 434.

[4] *Engineer*, vol. cxxix. (1920), pp. 107 and 132.

CHAPTER XII

POINTS IN THE DESIGN OF INTERNAL-COMBUSTION ENGINES

Attention has already been drawn to the complexity of problems confronting the designer of internal-combustion engines. He requires to have more than a working knowledge of such subjects as theory of machines and properties of materials, both physical and metallurgical, in order to design a machine which will run smoothly and not fail under the very considerable stresses to which it may be subject. He requires to study fuel technology in both its chemical and commercial aspects, so that he can arrive at a clear conception of the nature of his working substance and form a just appreciation of the supplies at his disposal. And he should be a fair mathematician, so that he can apply his knowledge to understanding and evolving the somewhat complicated thermodynamical relations which obtain. All this, of course, in addition to experience in works manufacture which has already been emphasised.

There has been a tendency in the past to treat the subject of internal-combustion engines too much in watertight compartments, with the result that certain types, such as Diesel oil engines, have often been considered in a class apart, and have, as a matter of fact, had more literature devoted exclusively to them than their inherent advantages deserve. It is true that the problem of the Diesel is a fascinating one, and has called forth the best faculties of the designer; but the emphasis laid on the type has tended to obscure other methods, such as sprayer oil engines, which are now proving at least as good in performance as Diesels, and which are so much simpler mechanically that they may very likely supplant the Diesel altogether.

The attempt, however, on the part of some authorities in recent papers to claim the sprayer oil engine as an offshoot of the Diesel by referring to them as Diesel engines with "solid injection" cannot be justified on theoretical grounds. If anything is typical of the Diesel it is the constant-pressure cycle under which it works. Sprayer oil engines are more akin to gas engines thermodynamically, in that they work on a modified constant-volume cycle or at most on a form of dual combustion.

Temperature Effects.—The whole question of internal-combustion engine design is closely bound up with the various effects which may be caused by the temperatures of the working substance. These effects fall roughly under four headings—

Expansion of the metals which form the walls of the combustion chamber.

Pre-ignition.

Evaporation of the lubricant of the piston.

Growth of cast iron at high temperatures.

The late Professor Bertram Hopkinson carried out experiments,[1] with the help of research students, on a Crossley gas engine with a single cylinder 11½ in. in diameter by 21 in. stroke, and rated at 40 B.H.P. when running at 180 R.P.M. The ratio of compression was about 6½ to 1. One of the objects of these experiments was to obtain definite knowledge about the temperatures of different portions of the combustion chamber.

For this purpose nickel-iron thermo-couples were fitted to the centre and lower edge of the piston, the centres of the exhaust and inlet valves, and later to a projecting bolt in the exhaust valve cover. It was found that under normal working conditions (about 10 per cent. gas) the temperature of the piston centre averaged about 380° C., whilst at the periphery it was only 210° C. For the heaviest charges used (11·0 per cent. gas) the temperature at the centre rose to 480° C. and at the edge 260° C. It is shown that the lower values stress the material about 5½ tons per sq. in., whilst the tension set up by the radiation effects of the heaviest charge reaches about 7½ tons per sq. in. These calculations assume the piston end to be a plain disc of uniform small thickness. The reinforcement at the edge by the piston trunk and any ribs would tend to reduce the above temperature stresses in practice. Hopkinson says, "It appears that the temperature at the centre of the piston will increase more or less as the square of the diameter if the thickness remains constant, or directly as the diameter if the thickness is increased in proportion thereto. Similar considerations apply to the valves."[2]

Under ordinary running conditions the temperature of the centre of the exhaust valve was found to be about 400° C. and at the centre of the inlet valve about 250° C. At a later stage of the experiments pre-ignition was obtained artificially by an iron bolt 4 in. long screwed into the exhaust-valve cover so that it projected into the centre of the compression space. Briefly, it was found that pre-ignition never occurred so long as the temperature at the tip of this bolt was under 690° C., but that it occurred every time if the temperature exceeded 740° C. The conclusion drawn is that a clean metal surface will ignite the gas when its temperature is a little above 700° C., but that pre-ignition will not occur if such a surface is below that temperature. It was further found that a temperature of 800° C. was sufficient to ignite the charge as soon as it came into the engine at atmospheric pressure.

From some indicator cards published with this paper it would appear that the average temperature of explosion in these experiments was about 1790° C., but these values were not directly measured.

This was, however, done by Professor E. G. Coker and W. A. Scoble[3] on a smaller gas engine built by the National Gas Engine

[1] "On Heat Flow and Temperature Distribution in the Gas Engine," *Proc. Inst. Civ. Eng.* vol. clxxvi. (1908–1909), p. 210.

[2] *Ibid.* p. 214.

[3] "Cyclical Changes of Temperature in a Gas-Engine Cylinder," *Proc. Inst. Civ. Eng.* vol. cxcvi. (1913–14), p. 1.

Co., which had a 7 in. diameter cylinder by 15 in. stroke, and developed 12 B.H.P. at 240 R.P.M. The ratio of compression was 5·8 to 1. In these experiments specially constructed thermo-couples of platinum-iridium and platinum-rhodium alloys were inserted into the combustion chamber from above and suitably protected from the violence of the explosion. The results show that with 12 per cent. ratio of gas to cylinder contents (average lower cal. val. 480 B.Th.U. per cub. ft.) the highest temperature of the explosive charge reached 1836° C. With 13 per cent. mixture this temperature rose to 1947° C., whilst with 15 per cent. mixture 2249° C. was recorded.

Although such high temperatures only occur at the peak of the indicator card, they mean that the contents of the combustion chamber must glow with an intense white heat periodically during each cycle. The radiation effects of such glowing gases, which vary theoretically as the fourth power of their absolute temperatures, play an important part in heating up the end of the piston and the walls of the combustion chamber. The water jacket is required just as much for removing this heat from the periphery of the piston end as for keeping the cylinder walls cool. The alternative, which is adopted in large-power gas engines and some high-speed Diesels, is to have water or oil-cooled pistons.

In sprayer oil engines a hot centre in the piston end is not altogether a disadvantage, since it tends to assist ignition if properly designed; but care should be taken to dispose of the mass of metal, so that temperature stresses between the centre and the rim are not excessive, whilst at the same time the temperature drop is sufficient, with the aid of the water jacket, to keep the rim below, say, 300° C.

If the temperature of the best mineral cylinder oils is allowed to exceed about 315° C. the lubricant evaporates rapidly and seizing of the piston may occur.

Growth of Cast Iron at high Temperatures.—The Phenomenon known to metallurgists as the growth of cast iron under temperature, is one which requires increasing attention from the internal-combustion engine designer.

Published information on the point is scattered over a number of papers,[1] but the experience of the various authors is not always in agreement, and it would therefore be advisible to consult a trained metallurgist for any important case.

Growth of cast iron amounting in extreme cases to more than 60 per cent. increase in volume takes place when the chemical composition of the metal is such that free carbon is formed under heat, and the gases penetrate into the casting by way of the graphite plates. It may

[1] Reference may be made to:

(*a*) "The Effect of Working Temperatures on Parts of Internal-Combustion Engines," Dr. C. H. Carpenter, *Proc. Staffordshire I. & S. Inst.*, Dec. 15, 1916, reprinted in Dr. W. H. Hatfield's book on "Cast Iron in the Light of Recent Research," 1918 ed., p. 149.

(*b*) "Cast Iron with Special Reference to Engine Cylinders," J. E. Hurst, *Manchester Ass. of Eng.*, Dec. 9, 1916.

(*c*) "Diesel Engine Castings," F. J. Cook, *North-East Coast Inst. of Eng. and Shipbuilders*, vol. xxxvi. (Feb. 1920).

(*d*) "The Metallurgy of Iron," Turner.

begin at temperatures of 250° C. Carpenter states that there are, no doubt, cases in which the physical influence of the expansion of dissolved gases in close grey irons makes itself felt to a considerable extent in creating a high internal pressure, which causes the iron to expand.

Mr. E. L. Rhead, the head of the Metallurgical Department of the Manchester College of Technology, suggests that a suitable mixture for pistons and cylinder liners can be obtained from: selected shop scrap, 50 per cent.; a good Mn. iron pig, 30 per cent.; mild steel scrap (such as boiler plate not less than $\frac{3}{8}$ in. thick), 20 per cent. Such a mixture should analyse as follows:—

Total carbon	content about	3·25	per cent.
Manganese	at least	0·7	,, ,,
Silicon	up to	1·2	,, ,,
Phosphorus	,, ,,	0·6	,, ,,
Sulphur	,, ,,	0·1	,, ,,

and have a bending strength of 36 cwt. It would not require annealing, but more care would be necessary in the melting and casting, as the shrinkage would be greater than with ordinary cast iron.

The presence of manganese appears to hinder the separation of the carbon and to toughen the alloy. The percentage addition of steel scrap depends to some extent on the analysis of the pig and iron scrap with which it is smelted. Its object is to lower the silicon and phosphorus contents and strengthen the casting. F. J. Cook (*c*) points out that for cylinder covers or breech ends the silicon may be increased to 1·5 per cent. and the manganese to 1·0 per cent. He also considers that more than 15 per cent. of mild steel will tend to make the metal too hard.

It is perhaps needless to point out that special care must be taken in the design of such castings, in order to avoid possible internal stresses set up by the cooling of the metal in the mould. The mass of metal should be disposed as evenly as possible and all re-entrant angles avoided by the use of large radii. The disposal of water pockets in the various positions of the jacket should not allow any possible sediment or steam to accumulate, and the whole water chamber should allow free positive circulation as much as possible.

Formation of the Working Substance.—A perusal of the various types of internal-combustion engines outlined in the last chapter will show what a fundamental part is played by the various methods of creating a suitable explosive mixture of air and combustible. It used to be said that nobody knew what was happening inside an internal-combustion engine cylinder at all parts of the working stroke. Nowadays the more important problem is to ascertain what is happening inside the combustion chamber during the formation of the working substance. Research and theory both show that the ideal condition consists of a thorough intermingling of air and fuel throughout the whole space. Such a result is more easily attained when both gas (or oil vapour) and air are drawn in together and compressed together, than when air alone is compressed and the fuel injected afterwards. One of the reasons for the success of the Diesel engine undoubtedly lies in the

intermingling effect of the air which is used to inject the oil. It is often said that the high efficiency of a Diesel is in no small measure due to the excess of air present in the combustion space, which ensures sufficient oxygen for complete and efficient combustion. This is true in so far as it attains the object stated above, namely, a satisfactory intermingling of air and fuel.

In large gas engines, over 1000 B.H.P., the modern tendency is towards the Nuremberg or four-cycle double-acting horizontal type. The practice of British and some German firms is to limit the mean effective pressure (on an I.H.P. basis) to round about 60 lb. per sq in. Such values are easily obtained from waste gases such as blast-furnace gas, provided the gas is thoroughly cleaned before use. The Cockerill Co., of Seraing, in Belgium, have recently introduced a new type of valve gear which exerts considerably more control over the proportions of air and gas admitted during different parts of the suction stroke, and governs by a variation of mixture strength in a constant compression at all loads. With this gear the mean effective pressure is raised up to an average of 82 lb. per sq. in. and in some cases (using coke-oven gas) to 110 lb. per sq. in. on an I.H.P. basis[1] At the same time the maximum explosion pressure is reduced some 25 per cent. A diagram of this valve gear is shown in Fig. 112.

The mushroom inlet valve at the top has a sleeve fixed to it, which acts like a piston valve in uncovering the gas and air inlets. Inside this sleeve is a lantern or second sleeve valve, which is moved up and down by two vertical rods, each side of the valve spindle, connected through a horizontal layshaft and rockers to the governor gear. The gas inlet being underneath, it follows that when this lantern is at the top, as shown in the drawing, the gas and air ports are both free to open in proportion to the lift of the main inlet valve. If the lantern is lowered by the governor it tends to restrict the gas inlet port until, when it reaches its lowest position, the gas supply is cut off altogether and only air is drawn into the piston In addition the valve is adjusted to allow fresh air alone to enter first and give an automatic scavenge, which improves the quality of the working substance by eliminating the remains of the products of combustion of the previous working stroke.

The formation of a homogeneous working substance is the chief problem to be solved in the sprayer type of oil engine.

The oil requires to be atomised sufficiently to ensure spontaneous and complete combustion, whilst at the same time it must have sufficient penetrating force to reach all parts of the compressed air in the combustion chamber. Failure in either of these two conditions leads to irregular or heavy consumptions, or what is worse, incomplete combustions with a smoky exhaust and a rapid deposition of carbon. In the absence of pronounced turbulence, the size and arrangement of holes in the sprayer nozzle should vary according to the shape of the combustion chamber. Other things being equal, such as length of hole and oil pressure, the larger hole will give greater penetration. The sizes, number, and angles of such holes would depend on the most

[1] For indicator cards, see *Engineer*, vol. cxxx. (1920), p. 497. Fig. 112 is from the same article.

suitable form of spray to penetrate as evenly as possible into all parts of the combustion space.

In practice the diameter (d) of the holes varies from $\frac{1}{32}$ in. to $\frac{1}{64}$ in. It is unadvisable to make their length more than $2d$. It will be found that increasing accuracy is necessary in their manufacture, as a variation

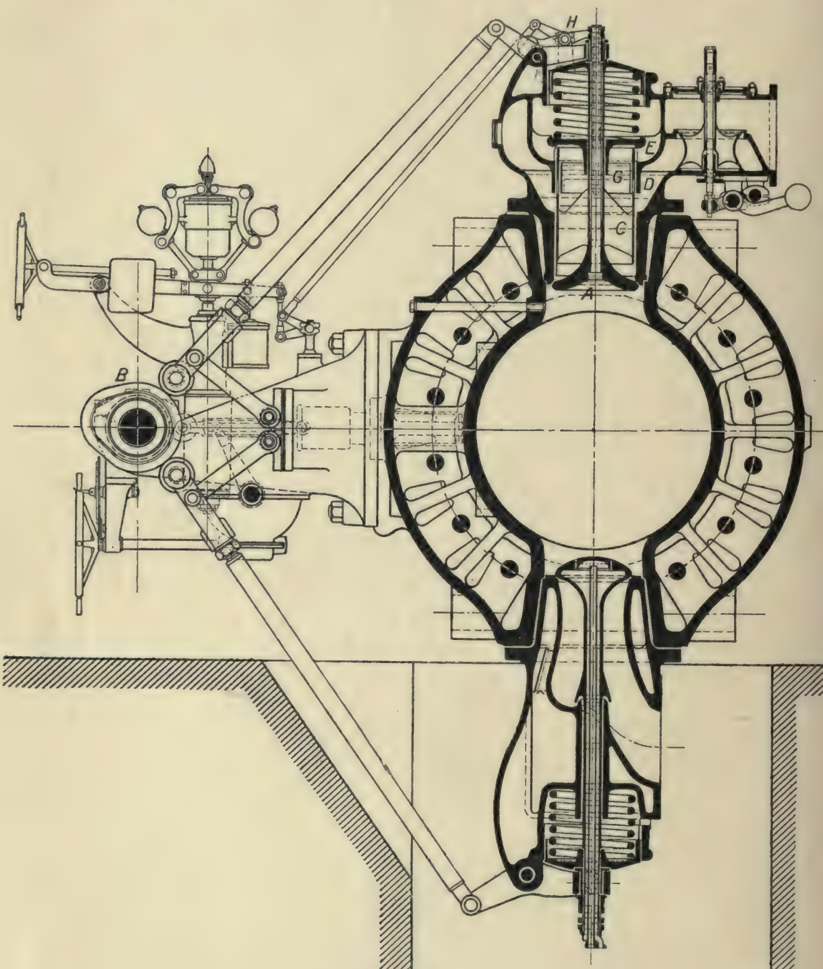


FIG. 112.—Valve gear for Cockerill gas engines.

of less than $\frac{1}{100}$ in. in the diameter may prove fatal. The spray should take the form of a fox's brush, and more consistent results would probably be obtained if the combustion space were shaped to the spray rather than the other way about.

The minuteness of the holes makes it imperative to strain the oil

well before entering the sprayer, as any small solid particle would upset the action of the nozzle.

Use of Indicator Cards in Preliminary Design.—The remarks which have been made about steam-engine diagrams (p. 179) apply with added force to the design of internal-combustion engines. The points of compression, explosion, and exhaust are not so easy to estimate, and the mean effective pressure from which to determine the proportions of the cylinder is usually assumed, unless actual indicator cards are available from old engines taken under similar conditions to those of the new design. The twisting-moment diagram, derived as it is from a combination of an indicator card and the inertia forces due to the reciprocating parts, is of more use to the design of internal-combustion engines than to steam. Engines working on the four-stroke cycle, for instance, have only one impulse in two revolutions, and the excess energy shown by the twisting-moment diagram is required to arrive at the correct proportions of the flywheel.

The Walker chart, already described on p. 201, enables a hypothetical indicator card to be constructed, which is sufficiently accurate for the purposes of preliminary design.

The information required or assumed is—

(*a*) Cycle of operation (*i.e.* constant volume, constant pressure, etc.).

(*b*) Ratio of compression (*alternatively* pressure at end of compression).

(*c*) The internal energy of the gas.

(*d*) Lower calorific value of the fuel in B.Th.U. per cub. ft. or lb., as the case may be.

(*e*) Ratio of air to fuel in the working substance.

(*a*) The choice of cycles of operation is usually limited in practice to constant-volume or to constant-pressure combustion, in both cases with compression-volume expansion. There are, however, engines which work on variable-volume combustion and a few which approximate to the dual-combustion cycle. All four types can be represented on the chart.

(*b*) The lowest PV curve on the chart (Fig. 113) is the compression curve of the desired indicator diagram. If the compression ratio (r) is known or assumed, the point of maximum compression is at once determined where this curve cuts the volume scale at the value $\frac{18}{r}$ cub. ft., *e.g.* for a compression ratio of 5·4 the volume at the end of compression would equal $\frac{18}{5 \cdot 4} = 3 \cdot 33$ cub. ft. and the pressure 135 lb. per sq. in. (abs.) from the chart point (1).

(The converse of this is often useful for obtaining an estimate of the ratio of compression from actual indicator cards, where the pressure at the end of compression can be measured.)

The temperature in ° C. (abs.), corresponding to a pressure of 135 lb. per sq. in. is obtained by striking horizontally from point (1) to the radial PT line, which represents a volume of 3·33 cub. ft., point (2). This reads 665° C. (abs.). This point can, if desired, be checked quite simply

from the relation $\frac{PV}{T}$ = constant. The chart is drawn for 1 lb. of gas occupying 18 cub. ft. at 373° C. (abs.) and 14·0 lb. per sq. in. Hence the required temperature

$$T = PV \times \frac{373}{14 \times 18} = 1{\cdot}48PV \text{ in } ^\circ\text{C. (abs.)}$$

(*c*) The internal energy of the mixture at the end of compression is then read off on the required curve at the temperature just obtained. This is done by dropping a line vertically from the point (2) to either the constant-pressure energy line if the cycle is constant pressure, or to

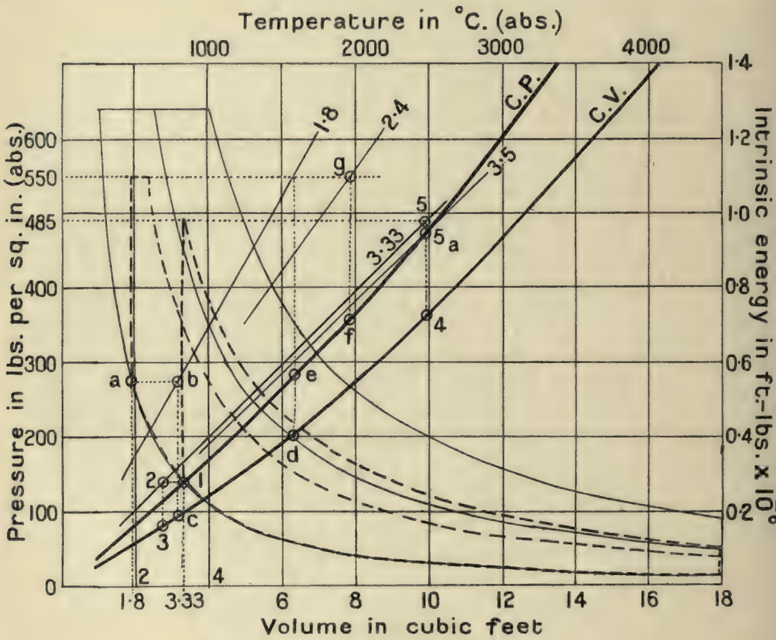


FIG. 113.—Applications of Walker's chart.

the internal-energy line if the cycle is constant volume, *e.g.* assuming the latter, the point (3) shows that the internal energy is $0{\cdot}16 \times 10^6$ ft.-lb. per lb.

(*d*) and (*e*) The lower calorific value of the fuel and the ratio of air to gas [1] are required in order to determine the heat supply in the mixture. This value, when added to the internal energy at the end of compression found under (*c*), gives the total energy in the gas at the end of combustion.

If the ratio of air to fuel is x to 1, than the calorific value of the mixture will equal the calorific value of the fuel divided by $x + 1$, *e.g.* when $x = 11{\cdot}32$ and the calorific value of the gas used is 500 B.Th.U.

[1] See p. 199 for computing the ratio of air to gas.

per cub. ft., then the mixture strength will equal $\frac{500}{12 \cdot 3} = 40 \cdot 6$ B.Th.U. per cub. ft.

As the chart is drawn up for ft.-lb. per lb., the heat supply in the working mixture equals

$$40 \cdot 6 \times 18 \times 778 = 0 \cdot 57 \times 10^6 \text{ ft.-lb. per lb.}$$

The total internal energy at the end of combustion then equals

$$(0 \cdot 57 + 0 \cdot 16) \times 10^6 = 0 \cdot 73 \times 10^6 \text{ ft.-lb. per lb.}$$

This point is shown as (4) on the internal-energy line of the chart.

Since in the example constant volume has been chosen the maximum pressure of explosion can be found by projecting vertically upwards from point (4) to the constant-volume line 3·33 cub. ft., point (5), and reading off the corresponding pressure on tne left-hand scale, *i.e.* 485 lb. per sq. in. (abs.).

If the variable-volume cycle had been worked to, and, for example, the volume at the end of explosion had increased to 3·5 cub. ft., it will be noted that the maximum pressure is reduced to 475 lb. per sq. in. (abs.) as shown by point (5*a*).

Walker's paper,[1] from which this example is taken, contains also examples of an air-cooled aero engine and, in the discussion, a Diesel engine. The card is best completed by laying a piece of tracing paper over the chart and following the expansion curves down to the exhaust pressure, as shown by the chain lines.

The Preliminary Design of a 400 H.P. Oil Engine.—As was the case with steam-engine design, information has to be obtained or assumptions made in order to work out the particulars of any required oil engine.

This information may be summarised as follows :—

(1) Type of engine.
(2) Nature of fuel.
(3) Normal duty and speed of engine.

The above are often given by the customer. The designer has then to select—

(4) A suitable compression ratio or alternatively a suitable compression pressure.
(5) The cycle of operation.
(6) The maximum explosion pressure, if it is desired to limit this to a given value.
(7) The ratio of fuel to air in the mixture.
(8) The ratio of $\frac{\text{B.H.P.}}{\text{I.H.P.}}$

For the purposes of an example it will be assumed that—

(1) The type of engine may be either a Diesel or a sprayer oil engine.

[1] *Proc. Inst. Mech. Eng.* Dec. 1920, pp. 1254 and 1305.

(2) The nature of the fuel will be crude oil with a lower calorific value of 18,000 B.Th.U. per lb., and the following approximate analysis :—

C	84·5 per cent.
H	12·5 „ „
O	2·0 „ „

(3) The engine is required to develop 400 B.H.P. when running at 200 R.P.M.

In this connection it is perhaps worth noting that the choice of speed is limited by the piston speed, the momentum and acceleration effects of the moving parts, and to some extent by the type of fuel. Light oils can run faster than crude oils, that is to say, petrol engines than Diesels. In practice these points give fairly wide ranges, but for oil engines the following figures may be taken as average values :—

Small petrol engines	1500 R.P.M.
Large petrol engines	900 „
Marine petrol engines . . .	700 „
Aero engines	1250 „
Diesel engines—	
Large marine	225 „
Submarine, large	450 „
„ small	600 „
Land Diesels	200 „
Hot-bulb engines	350 „
Sprayer oil engines	200 „

(4) *Compression ratio.*—The choice of a suitable compression ratio is chiefly a matter of experience. In the absence of more specific data the following table may serve as a guide :—

Normal oil and gas engines	$4\frac{1}{2}$–7
Petrol engines (car)	4–$4\frac{1}{2}$
„ „ (aero)	5–6
Sprayer oil engines	9–10
Diesel engines	12–16

For the example the Diesel will be taken as 14·5 (see p. 251) and the sprayer oil engine as 10.

(5) The choice of type practically limits the cycle of operation to four-stroke or two-stroke. In the example the four-stroke type will be adopted. The Diesel will work on the constant-pressure cycle and the sprayer oil on the dual-combustion cycle.

(6) The maximum explosion pressure of the Diesel will be limited to 500 lb. per sq. in. (abs.) and the sprayer oil engine to 550 lb. per sq. in. (abs.). This can be done in practice by controlling the injection of the fuel so that there is a lag towards the end of the injection.

(7) *Ratio of air to fuel.*—Using Moore's formula (p. 199)—

$$A = 0{\cdot}116(C + 3[H - \tfrac{1}{8}O])\text{ lb.}$$

$$A = 0{\cdot}116\left(84{\cdot}5 + 3\left[12{\cdot}5 - \frac{0{\cdot}2}{8}\right]\right)\text{ lb.}$$

$$= 14\text{ lb. of air to 1 lb. of fuel (theoretical).}$$

In practice the Diesel engine would use about twice as much air again, or say 28 lb. to 1 lb. of fuel. The sprayer oil engine, on the other hand, may be given $2\frac{1}{2}$ times the theoretical quantity, or say 36 lb. per lb. of fuel.

(8) The mechanical efficiency of a Diesel, including the air-compressor drive, may be taken at 0·75, and the sprayer oil engine at 0·85, as a conservative estimate.

Indicator Diagrams from Walker's Chart.

Diesel Engine.—Assuming a compression pressure of 500 lb. per sq. in. (abs.) Walker's chart, p. 201, following the lowest PV curve, shows that 18 cub. ft. is compressed to 1·24 cub. ft., or a compression ratio of $\frac{18}{1 \cdot 24} = 14 \cdot 5$. Following the 500-lb. pressure line horizontally till it strikes the radial PT line, corresponding to 1·24, gives the temperature (on the top scale) and the corresponding energy vertically below on the *constant-pressure* energy line (reading on the right-hand scale). This is found to be $0 \cdot 325 \times 10^6$ ft.-lb. per lb. The heat energy per lb. of fuel

$$\frac{\text{cal. val.}}{x+1} \times 778 = \frac{18{,}000}{28+1} \times 778 = 0 \cdot 483 \times 10^6 \text{ ft.-lb.}$$

When combustion is complete the energy would be the sum of these two quantities, or $0 \cdot 808 \times 10^6$ ft.-lb. Finding this point on the constant-pressure energy line and moving vertically upwards (*i.e.* keeping to the same temperature line) to 500 lb. per sq. in. shows the volume at the end of combustion would be practically 2·9 cub. ft.

If a piece of tracing paper is now laid over the chart the indicator card can be drawn by following an intermediate expansion curve down to 18·0 cub. ft. from this volume of 2·9 cub. ft. on the 500-lb. pressure line. From this card the theoretical indicated M.E.P. is found to be 102 lb. per sq. in., or multiplying by the mechanical efficiency the theoretical brake M.E.P. $= 102 \times 0 \cdot 75 = 76 \cdot 5$ lb. per sq. in.

Sprayer Oil Engine.—As this is an example of the dual-combination cycle the values worked out here have been added to Fig. 113 (p. 248) in chain-dot lines and small lettering.

Assuming a compression ratio of 10 the final compression volume will be $\frac{18}{10} = 1 \cdot 8$ cub. ft., corresponding to a compression pressure of 285 lb. per sq. in. (abs.) (*a*).

Moving horizontally to this volume on the radial PT line 1·8 (*b*) and then dropping vertically to the internal-energy curve gives a value for this latter of $0 \cdot 19 \times 10^6$ ft.-lbs. (*c*).

The heat energy per lb. of the fuel with an air ratio of 36 to 1

$$= \frac{18{,}000}{36+1} \times 778 = 0 \cdot 378 \times 10^6 \text{ ft.-lbs.}$$

Hence the total corresponding energy

$$= (0 \cdot 19 + 0 \cdot 378)10^6 = 0 \cdot 568 \times 10^6 \text{ ft.-lbs.}$$

If all the fuel were burnt at constant volume the pressure would rise to over 700 lb. per sq. in. (abs.), but actually the oil injection would be controlled to give a maximum explosion pressure of 550 lb. per sq. in. (abs.).

Dropping down from the PT line for 1·8 at this pressure of 550 lb. to the internal-energy curve shows that the value would be $0{\cdot}41 \times 10^6$ ft.-lb. (*d*), leaving $(0{\cdot}568 - 0{\cdot}38)10^6 = 0{\cdot}188 \times 10^6$ ft.-lbs. to be utilised at constant pressure. The energy on the constant-pressure line corresponding to $0{\cdot}41 \times 10^6$ ft.-lb. of internal energy is $0{\cdot}56 \times 10^6$ ft.-lb. (*e*). Hence the energy at the end of combustion on the constant-pressure line is $(0{\cdot}53 + 0{\cdot}188)10^6 = 0{\cdot}718 \times 10^6$ ft./lb. This point (*f*) shows vertically above it at 550 lb. point (*g*) that the volume (from the PT line) has been increased to 2·4 cub. ft. The resulting indicator card is then drawn in as shown by the chain-dot lines. From this the indicated M.E.P. is 84 and the brake M.E.P. $84 \times 0{\cdot}85 = 71{\cdot}3$ lb. per sq. in.

Cylinder Proportions.—It is now possible to compute the volume of the cylinders in the usual way from the general relation

$$\text{I.H.P} = \frac{\text{P.L.A.N.}n}{33{,}000}$$

where P = the indicated M.E.P. in lb. per sq. in.

$$\text{L} = \text{stroke in ft.} = \frac{l}{12}, \text{ where } l = \text{stroke in in.}$$

a = area of cylinder in sq. in.

N = number of impulses per minute

$$= \frac{\text{R.P.M.}}{2} \text{ for four-stroke cycle}$$

n = number of cylinders

$$\text{I.H.P.} = \frac{\text{B.H.P.}}{\eta}, \text{ where } \eta = \text{the mechanical efficiency}$$

400 B.H.P. could be divided between either three or four cylinders for both types of engine under discussion. Most firms at the present time would probably use four cylinders, chiefly on account of the better balancing effects, though the engine would be more expensive on that account. This would give a cubic capacity per cylinder of

$$la = \frac{\text{B.H.P.} \times 33{,}000 \times 12}{\eta \times \text{N} \times n \times \text{P}}$$

$$= \frac{400 \times 33{,}000 \times 12}{0{\cdot}75 \times 100 \times 4 \times 102} = 5180 \text{ cub. in.}$$

for the Diesel,

or
$$= \frac{400 \times 33{,}000 \times 12}{0{\cdot}85 \times 100 \times 4 \times 84} = 5540 \text{ cub. in.}$$

for the sprayer oil engine.

The choice of stroke to bore ratio can best be determined from

previous experience and from an investigation of the practice used in the most successful engines. Tables showing such practice for marine Diesel engines are given in *Engineering*, vol. xcvi. (1913), p. 431, and in *Engineering*, vol. cx. (1920), p. 590. Both these sets of tables also show piston speeds, R.P.M., cylinder dimensions, and the M.E.P. on both an indicated and a brake basis. The question has been studied by W. Stremme, of Budapest,[1] who makes comparisons between the most economical of the short and long-stroke Diesels when considered on a basis of consumption, initial cost, and reliability. It is shown that the long stroke $\left(\frac{1\cdot5}{1}\right)$ is the most reliable, as it has a lesser consumption, which is attributed to a better mixture, and that it requires less lubicating oil, due to a smaller bearing pressure. Its capacity for overload is at least 7 per cent. greater, other things being equal. It is, however, heavier and more costly to build. Short-stroke engines $\left(\frac{1}{1}\right)$ are stated to be more suitable for plants where the capital cost is to be kept low compared to depreciation and interest on capital. This applies especially to standbyes and other installations with a low load factor. For motive power (*i.e.* marine practice) as opposed to generating power the long-stroke engine is much the best, on the score of greater reliability and flexibility. The ratio $\frac{\text{stroke}}{\text{bore}}$ should not in this case be less than 1·3 and may be as high as 1·9.

For sprayer oil engines experience has not been sufficient to indicate the best stroke to bore ratios for every given set of conditions. A ratio of $\frac{1\cdot5}{1}$ is frequently used at present.

Choosing this ratio for both the Diesel and the sprayer oil engine of the present example gives when $\frac{l}{d} = 1\cdot5$

$$\text{cylinder volume} = la = \frac{l\pi d^2}{4} = \frac{1\cdot5\pi d^3}{4}$$

whence for Diesel $d^3 = \frac{4}{1\cdot5\pi} \times 5180 = 4400$

$d = 16\cdot375$, say $16\frac{3}{8}$ in. diameter

and $l = 24\frac{1}{2}$ in.

For sprayer oil engine $d^3 = \frac{4}{1\cdot5\,\pi} \times 5540 = 4690$

$d = 16\cdot72$, say $16\frac{3}{4}$ in. diameter.

and $l = 25$ in.

[1] *Zeitschrift des Vereines deutscher Ingenieure*, vol. lx. (1916), pp. 561–565 and 588–591. See also *Ibid.* vol. lxi. (1917), pp. 224–227.

Summarising the design gives

400 B.H.P. oil engine 200 R.P.M.

four cylinders four-stroke cycle working on crude oil with a lower calorific value of 18,000 B.Th.U.

	Diesel.	Sprayer oil engine.	Units.
Compression ratio	14·5	10	—
Type of combustion cycle . .	constant pressure	dual combustion	—
Maximum pressure	500	550	lb. per sq. in. (abs)
Ratio of air to oil	28 : 1	36 : 1	—
Mechanical efficiency . : . .	0·75	0·85	$\frac{\text{B.H.P.}}{\text{I.H.P.}}$
Indicated mean effective pressure	102	84	lb. per sq. in.
Cylinder volume	5180	5540	cub. in.
Ratio of stroke to bore . . .	1·5 : 1	1·5 : 1	—
Cylinder Diameter	$16\frac{3}{8}$	$16\frac{3}{4}$	in.
Stroke	$24\frac{1}{2}$	25	in.
Piston speed (2L × R.P.M.) .	816	833	ft. per min.

INDEX TO TEXT

[*The numbers refer to pages.*]

E

F

G

S 2

U

V

THE END.

PRINTED IN ENGLAND BY WILLIAM CLOWES AND SONS, LIMITED, BECCLES.